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Tank Thermal Response Analysis, Phase II

NUMERICAL MODELING OF A TANK CAR UNDER FIRE CONDITIONS

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17. Abstract

A modelling approach for the thermal response of LPG tanks exposed to fire conditions is reported. The step-by-step application of the technique, which was previously used with some success for the modelling of reactor pressurizers, is given in detail along with the proposed phenomenological relations. The technique requires that the tank contents be divided into two control volumes separated by an interfacial region. The bottom control volume contains subcooled or two-phase vapour. Heat is transferred into both regions from the flame through the shell and into the lading. There is also a mass and energy flow from the vapour region through the pressure relief valve when the internal pressure exceeds the pressure set of the valve. Mass and energy exchange across the interface occurs in the form of evaporation or bubble rise from the bottom control volume to the top control volume. Conservation equations of mass and energy for each control volume are coupled with the lading equations of state to yield an explicit solution for the thermal response of the tank vessel. The results of the model are compared to field tests conducted on one-fifth scaled tanks with reasonable agreement.

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17. Résumé

Ce rapport présente l'approche utilisée en vue de modéliser le comportement thermique de réservoirs de GPL soumis à l'action des flammes. Il décrit étape par étape l'application de cette technique jusqu'ici utilisée avec un certain succès pour modéliser les pressuriseurs de réacteurs nucléaires, ainsi que les relations phénoménologiques en jeu. Cette technique requiert que le réservoir soit divisé en deux volumes de contrôle séparés par une interface commune, le volume inférieur renfermant de la vapeur sous-refroidie, ou sursaturée. La chaleur est transférée aux deux volumes à travers les parois. Il existe, en outre, un transfert de masse et d'énergie depuis l'espace vide vers l'extérieur, par l'entremise de la soupape lorsque celle-ci s'ouvre sous l'action d'une pression supérieure à la pression de consigne. Les transferts de masse et d'énergie à travers la région interfacielle se produisent sous forme de mouvement ascensionnel des bulles ou d'évaporation. Les équations de conservation de masse et d'énergie à l'égard de chaque volume de contrôle sont combinées aux équations d'état de la cargaison, débouchant sur une solution explicite rendant compte du comportement thermique du réservoir. Les résultats théoriques ainsi obtenus concordent assez bien avec ceux des essais conduits sur une maquette au 1/5e d'un réservoir.

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SUMMARY

This report presents a numerical model of the fire-induced heat transfer to and thermohydraulic response of a pressure liquefied gas contained in a horizontal cylinder. The work was undertaken by the Fire Science Centre of the University of New Brunswick as part of a research program concerned with the safe transport of flammable pressure liquefied gas. The work was supported by Transportation Development Centre (TDC) through DSS File No. 11SD.T8200-5-5517.

The derived model assumes that the tank consists of vapour and liquid regions separated by an interface and heated externally around its circumference. It is further assumed that:

i) the only modes of energy and mass transfer across the interface are due to evaporation and bubble rise;

ii) bubble rise only occurs when the lower control volume is in a two-phase state (i.e., saturated conditions prevail);

iii) bubbles leaving the liquid region do so with the saturated vapour enthalpy corresponding to the vessel pressure;

iv) the bubbles reach terminal velocity instantaneously;

v) Harmathy's correlation for terminal bubble rise velocity is applicable to this geometry;

vi) mass leaving the bottom control volume through evaporation absorbs the latent heat of evaporation corresponding to the vessel pressure from the liquid region;

vii) mass entering the top control volume through evaporation does so with the saturated vapour enthalpy corresponding to the vessel pressure;

viii) condensation on the internal tank shell and the bulk vapour region is neglected;

ix) over a time step, constant values for properties and parameters are considered throughout each control volume;

x) pressure is constant throughout the tank car over each time step:

xi) the mass flow through the pressure relief value is considered to follow a polytropic process; and

xii) the volume of the tank car remains constant.

The coupled conservation equations of mass, energy and volume plus the equations of state for the two control volumes are coupled together in order to obtain an explicit equation for the time derivative of pressure (dP/dt). Once pressure is determined, the other time-dependent derivatives of properties can be obtained; thus, given initial conditions at time "t", the new conditions at time "t + dt" can be calculated. Four state conditions within the tank are possible with the model:

a) vapour superheated, liquid subcooled;

b) vapour superheated, liquid saturated;

c) vapour saturated, liquid subcooled; and

d) vapour saturated, liquid saturated.

The governing equations for each possible condition are unique and are elaborated in the report. The numerical solution is obtained by combining these equations with criteria for the determination of state and phenomenlogical relations for the fire heat input (radiation and convection), internal heat transfer coefficients, vessel wall transient conduction, evaporation, bubble rise, and pressure relief valve mass flow. For each condition and region, its state and quality can be determined and the derivative of pressure with respect to time obtained explicitly. This also leads to the explicit calculation of the time derivatives of liquid and vapour enthalpy resulting in the determination of the time derivatives of mass. The values of enthalpy, pressure and mass are then updated, through integration and the procedure is repeated for each time step.

The numerical simulation was validated with the experimental results of an unprotected 1/5 linear scale propane-filled tank. In order to test the performance of the model, numerical experiments were performed where the fire input and the vapour generation were systematically varied.

SOMMAIRE

Le present rapport expose un modele numerique du transfert, a un gaz liquefie sous pression dans un cylindre horizontal, de la chaleur produite par une flamme, et de la reponse thermo-hydraulique de ce gaz. Ce modele a ete elabore par le Fire Science Centre de l'Universite du Nouveau-Brunswick dans le cadre d'un programme de recherche sur las securite du transport de gaz flammables liquefies sous pression. Les travaux ont ete parraines par le Centre de developpement des transports (MAS-dossier 11SD.T8200-5-5517).

Le modele suppose que le reservoir contient de la vapeur et du liquide separes par une interface et qu'il est chauffe exterieurement sur sa circonference. Autres hypotheses:

i) les transferts d'energie et de matiere a travers l'interface ne peuvent se faire que par evaporation ou montee de bulles;

ii) les montees de bulles ne peuvent survenir que lorsque le volume de controle inferieur est dans un etat a deux phases (c'est-a-dire lorsque la vapeur est saturee);

iii) les bulles quittent la region liquide lorsque l'enthalpie de la vapeur saturee correspond a la pression dans le reservoir;

iv) les bulles atteignent leur vitesse limite instantanement;

v) la correlation de Harmathy pour la vitesse de montee limite des bulles est applicable a cette geometrie;

vi) la masse qui s'echappe du volume de controle inferieur par evaporation absorbe la chaleur latente d'evaporation correspondant a la pression du liquide dans le reservoir;

vii) durant le transfert de masse as volume de controle superieur par evaporation, l'enthalpie de la vapeur saturee correspond a la pression dans le reservoir;

viii) la condensation sur la paroi interieure du reservoir et au sein de la vapeur est negligeable;

ix) durant chaque increment de temps, les grandeurs et les parametres ont des valeurs constantes dans chaque volume de controle;

x) durant chaque increment de temps, la pression est uniforme dans tout le wagon-citerne;

xi) l'ecoulement dans la soupape de surete est un processus polytropique; et

xii) le volume du wagon-citerne est constant.

En combinant les equations couplees de conservation de la masse, de l'energie et du volume avec les equations d'etat pour les deux volumes de controle, on obtient une expression pour la derivee de las pression par rapport au tempts (dP/dt). Une fois la pression connue, les derivees par rapport au temps des autres grandeurs, qui varient avec le temps, peuvent etre obtenues; ainsi, connaissant les conditions a l'instant "t", on peut calculer les conditions a l'instant "t + dt". Quatre situations sont possibles dans le reservoir avec ce modele:

a) vapeur surchauffee, liquide sous-refroidi;

b) vapeur surchauffee, liquide sature;

c) vapeur saturee, liquide sous-refroidi; et

d) vapeur saturee, liquide sature.

Chacune de ces situations a ses propres equations determinantes, lesquelles sont donnees dans le rapport. Ces equations permettent d'obtenir une solution numerique lorsqu-elles sont combinees aux criteres utilises pour determiner les relations d'etat et les relations phenomenologiques pour la chaleur transmise par la flamme (rayonnement et convection), les coefficients de transfert de chaleur interne (dans la vapeur - rayonnement et convection - et dans le liquide - convection et ebullition, en conditions de sous-refroidissement et de saturation), la conduction transitoire dans la paroi du reservoir, l'evaporation, la montee des bulles et l'ecoulement dans la soupape de surete. Pour chaque situation et chaque region, on peut determiner l'etat du systeme et obtenir une expression explicite pour la derivee de la pression par rapport au temps. Ceci permet egalement d'obtenir des expressions explicites pour les derivees par rapport au temps de l'enthalpie du liquid et de la vapeur et, part consequent, de determiner la derivee de la masse par rapport au temps. Les valeurs de l'enthalpie, de la pression et de la masse sont ensuite corrigees par integration et l'operation est repetee a chaque increment de temps.

Les resultats obtenus avec le modele numerique ont ete confirmes par des experiences effectuees sur un reservoir non protege a echelle lineaire 1/5 rempli de propane. Dans ces comparaisons, on a fait varier systematiquement la quantite de chaleur fournie par la flamme et la production de vapeur.

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NOMENCLATURE

Symbols

A	area
c	specific heat
c_p	specific heat at constant pressure
Cd	valve discharge coefficient
d	bubble diameter
g	acceleartion of gravity
h	specific enthalpy
h_{ev}	interfacial evaporative heat transfer coefficient
h_{fg}	specific enthalpy of vaporization
k	thermal conductivity
L_B	mean bubble path (wall-to-interface)
m	mass
(mh)	rate of energy transfer to a control volume
Nu	Nusselt number
P	pressure inside the tank car
Pr	Prandtl number
q/A	heat flux
Q'	heat flow rate into a control volume
r	radial position
R	gas constant
Ra	Rayleigh number
t	time
T	temperature
THFM	total maximum heat flux
v	specific volume
v_{fg}	difference between the specific volumes of saturated vapour and of saturated liquid (v_{gs} - v_{fs})
V	volume
V_H	rate of change of volume in stratified layer
V_t	bubble rise terminal velocity
x	vertical position in stratified layer
XB	quality of the liquid region
XT	quality of the vapour region

Greek letters

α	angle measured in the anticlockwise direction from top vertical
β	thermal coefficient of expansion
γ	specific heat ratio
σ	thickness of stratified layer
Δ	change
Θ	angle from top dead centre of the tank shell in rads
μ	dynamic viscosity
ρ	density
σ	surface tension

Subscripts

bub	related with bubble rise
B	bottom control volume
ev	related to evaporation
fs	saturated liquid
F	liquid space
Ff	liquid in liquid space
Fg	vapour in liquid space
Fi	net rate into liquid space
gs	saturated vapour
G	vapour space
Gf	liquid in vapour space
Gg	vapour in vapour space
Gi	net rate into vapour space
re	relief valve
st	stratified conditions

Superscripts

	time derivative

INTRODUCTION

Tanks laden with hazardous fluids, such as propane, butadiene, ammonia, ethylene, vinyl-chloride and chlorine, have been involved in a number of accidents in recent years. These accidents have caused death, serious injury, extensive property damage, and disruptive evacuations (1).

In a typical railway accident, a tank car carrying a compressed flammable commodity such as propane is punctured by impact. The propane is released then ignited by sparks or a hot car component. A poolr or torch fire results which may engulf or impinge upon other tank cars. If the propane is not properly vented by a suitable pressure relief valve the tankc ar may fail and explode (1).

An understanding of the processes and phenomena involved is of great importance before preventive measures can be undertaken. Scaled and full-scale tests have been performed on instrumented cars in an attempt to understand such processes and phenomena. This descrutive testing is very expensive and provides valuable data. However, a complete assessment of the intervening parameters is nearly impossible due to a number of factors, such as wind, ambient temperature and faulty or insufficient instrumentation.

A simulation model, properly benchmarked using existing field data, will enhance our understanding of the processes involved and will be an effective and powerful design tool.

In this report, a simulation model for the tank car is developed by extending the pressurizer modeling approach proposed by Nahavandi and Makkenchery (2) and Baron (3). The model describes the time-dependent heat flux into the tank car and its lading, the time-dependent pressure history of the tank contents and the time-dependent mass discharge through the pressure relief valve during pool fire engulfment and torch fire impingement.

LITERATURE SURVEY

The literature survey will be divided into three major sections: i) materials relevant to tank car modeling; ii) existing models of tank cars subjected to a fire environment, and iii) experimental results of field tests.

MATERIALS RELEVANT TO TANK CAR MODELING

Two-Phase Flow Models

Nahavandi and Makkenchery (2) and Baron (3) described an approach dealing with the modeling of typical pressurizers of light water reactors (PWR) which allows a considerable degree of flexibility in the possible number of thermodynamic processes and states. No "a priori" assumptions are required for the states and their determination is just a function of heat input.

The main features of this method are as follows: The pressurizer contents are made up of two distinct control volumes separated by a clearly defined interface. The contents of these control volumes have their own properties and each control volume is thermodynamically analyzed through energy, mass and volume balances. The interaction between the two phases occurs through condensate drop and bubble rise. The equations for each phase are combined with the equations of state and reduced to yield an explicit differential equation for pressure with respect to time. Once pressure is known, values for volume, mass, enthalpy and other properties are determined. Therefore, given the geometry of the vessel and the state of the system at time "t", the model can determine parameters and properties at time "t + dt". A particular difficulty with this approach is the considerable complexity of the equations and the sensitivity of the code to some critical derivatives.

Goemans (4) also modeled a pressurizer but with a more detailed approach. He considered the pressurizer contents to be made up of three distinct regions: the steam volume, the main water volume and a buffer volume. The steam and main water volumes are coupled through energy, mass and volume balances while the buffer volume is coupled to the other two regions only through a volume balance. Some sources of energy and mass interchange between the regions investigated are of potential interest to the tank car model such as condensation of steam, flashing of water, relief valve flow and heat conduction through the vessel walls. This model only considers the vapour region to be either a dry saturated or superheated vapour.

Chiu and Poon (5) performed a dynamic analysis of a pool boiling plant by using a small perturbation technique which is essentially valid except at startup and shutdown operations where the perturbation is much too great. In using this technique, nonlinear differential equations are reduced to linear form. In the experimental work it was decided that frequency response tests would give more complete information than transient response tests. Factors considered in this model include: i) steam voids in the water phase, ii) water drops in the steam phase, and iii) steam control valve characteristics. This report illustrates some of the interesting bubble dynamic relationships including a relationship for mass flow rate of bubble rise which is independent of vessel geometry. The valve characteristic model presented is a simplified, linearized model relying heavily on extensive experimentation.

Lin et al. (6) presented the RELAP5/MOD2 model for describing repressurization transients and report on the results obtained through simulation of three separate experiments. RELAP5/MOD2 is a new improved version of RELAP5 (7,8) thermal hydraulic computer code used to model pressurized water reactors. The most notable improvement is the capability to model coexisting subcooled liquid and superheated vapour. In the development of the model, the hydrodynamic and thermodynamic process models are based on first principles of the physical phenomenon observed in representative experiments. Three separate effects have been used to assess RELAP5/MOD2. The results present good agreement with the experiments, an indication that the technical approach used as a good physical background.

Moody (9) presented a general analytical model for predicting liquid/vapour action in an arbitrarily-shaped vessel during blowdown, including the effects of local heating and mass flows. This model is used to generate time dependent mixture levels and local mixture states during blowdown from an adiabatic, right cylindrical vessel at 1000 psia. The model considers the contents of the vessel to be at saturated conditions. A bubble rise velocity is assumed constant throughout the whole process. Calculation is performed through a number of nodes in the vertical direction. At each nodal level, the complete cross-section is considered to have constant properties including mixture, while pressure is considered uniform throughout the vessel at any one time. Predicted mixture levels are in reasonable agreement with experimental values for vapour blowdown tests.

Hutcherson et al. (10) considered the decompression of an initially-saturated liquid. The decompression process is broken up into three major regimes: i) an initial subcooled regime, ii) a bubble growth regime, and iii) a subsequent dispersed regime. The subcooled analysis considers the elasticity of both the liquid and the vessel. This regime is assumed to prevail until dissolved noncondensable gases come out of the fluid. The bubble growth analysis is made for thermally dominated bubble growth from a solid surface to a superheated liquid. Inertially-dominated bubble growth occurs, but since its period is very short, it has been neglected in the analysis. Correlations for bubble growth and departure velocity are presented. The net effect of the volume production within and the volume escape rate from the system is then assumed to be the controlling mechanism of the system pressure. The dispersed analysis assumes that the vapour bubbles have grown sufficiently so the liquid is uniformly distributed within the vapour phase. The transition is observed to occur at a void fraction of approximately 0.3, then the remaining fluid is assumed to expand isentropically.

The ratio of the number of nucleation sites to the wetted surface area was determined to be a critical factor in the pressure history. As the number of potential nucleation sites increases, so, too, does the predicted pressure level.

Agee et al. (11) reported on a method to obtain the thermodynamic pressure from the equation of state for fluids under equilibrium or nonequilibrium conditions. The method is applicable to models of compressible two-phase flow that account for the temperature of each phase in the mixture. The formulation can also handle such cases as flashing of superheated liquid and condensation of subcooled vapour in which metastable thermodynamic states are attained. The state properties for a phase in a metastable state are not available but extrapolation of the equations of state are assumed to be correct.

Mathematical models of two-phase flows require mass, energy and momentum equations for each phase present in the flow field. Coupling of such equations with the equation of state closes the system of equations to enable a solution. Therefore, pressure must be expressed as a function of mass and energy for each phase and the derivatives of pressure with respect to mass and energy must also be obtained through the use of implicit functions.

The work of Wang et al. (12) dealt with a new approach to numerically simulate the steam-water two-phase heat transfer process. In the past, work in areas related with nucleate boiling was mostly empirical, with no attempt to describe the fundamental nature of the process. The approach followed in (12) emphasized the phenomenology of the process and its essence is based on keeping track of the microscopic boiling events and the macroscopic phenomena will emerge as a natural consequence.

Bubble growth relationships are presented for bubbles which are stationary, or in a laminar or turbulent flow regime. The paper also presents an interesting method to determine the locations of bubble nucleation sites. Whenever mean nucleation site density is given, a random number generator and a Poisson function are used to determine the locations. A correlation for departure velocity, in addition to superheated and subcooled forced convection boiling simulation, are also presented.

Valve Modeling

Jones and Underwood (13) critically analyzed formulations used to calculate the release rate of pressure liquefied gases. Expressions are presented which evaluate one-phase (liquid) flow or two-phase flow. Two theories for two-phase flow are presented; homogeneous flow and annular flow and the two resulting expressions are discussed and compared.

The variation of the modified discharge coefficient (CDM) with l/d is presented where l/d is the ratio of length to diameter of the outlet horizontal pipe. It was found that the homogeneous equation did not correlate well with the available data and in most cases, CDM exceeded unity. The annular flow theory produced a consistent variation of CDM with l/d regardless of the flow medium and the extent of superheat. Although CDM exceeds unity for $l/d \leq 15$, the results show a substantial improvement over those resulting from homogeneous flow theory. In their work, the criterion for the initiation of two-phase flow with superheated liquids is taken to be 1/d > 3; however, they show that this criterion does not apply to all superheated liquids.

Friedel and Lohr (14) described a general calculation procedure for sizing cross-sectional areas of pressure-relief devices and relief line components in two-phase flow. In setting up the procedure, the phenomena and changes of state in the pressure vessel and the pressure drop characteristic of the relief line during two-phase flow are first considered with the help of equations of conservation, so that the state in the container provides the initial parameter of the efflux problem. For a given relief line, the calculation produces the unsteady efflux of mass after the response of the pressure relief device, and hence the variation of pressure with time in the reaction vessel.

According to a recent study (15), spring-loaded pressure relief valves open at the discharge pressure set, achieve full lift at approximately 103 percent of the opening pressure and rated flow at 100 percent of the opening pressure set. They begin to close at the closing set pressure but they do not reset until the pressure has dropped below 85 to 95 percent of the value of the closing pressure.

Flame Modeling

Birk and Oosthuizen (16) presented a computer model for the radiation heat transfer to a railway tank car from a pool fire surrounding the tank. The model assumes an infinitely long horizontal cylinder surrounded by radiating combustion gases bounded by an infinitely long rectangular domain. The radiating properties of the hot combustible gases can be represented by an effective flame temperature and absorption coefficient. Since reliable measurements of radiant heat transfer were not taken in either of the fire tests considered, there are no means of accurately verifying the predictions of the model. In the present study, these results will be used to generate the radiative heat flux as a function of temperature, absorption coefficient and angular position around the tank.

Tunc and Venart (17) evaluated the incident radiation on a horizontal cylinder from a pool fire and from a flare (18). Unlike Birk and Oosthuizen's model (16), which considers only circumferential variation of heat flux, Tunc and Venart's model considers both circumferential and longtudinal variations of heat flux on the cylinder surface. The effect of wind velocity on the pool fire and the flare with the resulting flame inclination on the incident radiative heat flux, is also investigated. Assumptions such as constant flame absorption coefficient and constant flame temperature throughout the flame volume are made in both (16) and (17). Tunc and Venart fit a fourth order polynomial to describe the flame shape as a function of flame height based on photographs, where Birk and Oosthuizen assumed a rectangular flame shape enclosing the actual flame volume. Tunc and Venart stated that the flame shape is an important parameter and controls the radiative heat flux coming to the cylinder surface.

The radiant heat flux from the flare is considered as an additional source of radiation on the tank car. The study investigates the dimensions of vertical and bent-over flares under crosswind conditions. The radiant heat flux is found to be strongly dependent on the inclination angle and the shape of the flare. It was found that radiant heat flux on the cylinder increases considerably with an increase in inclination angle. The representation of the flame contour also has a marked effect on the heat flux. By studying the flame photographs, a conical representation is more realistic than a cylindrical representation. The results of both the pool fire and the flare are compared to existing experimental results and good agreement is obtained.

Heat Transfer Coefficients

Lloyd and Moran (19) studied experimentally natural convection from horizontal surfaces of circular, square, rectangular and right triangular platforms. Electrochemical techniques were employed to determine the natural convection through analogous mass transfer measurements. The Schmidt numbers were in the order of 2200 so that these results apply to the analogous high Prandtl number heat transfer problem. The results encompass a wide range of Rayleigh numbers (Ra) covering both the laminar and turbulent regimes. Transition was found to occur at about $Ra = 8 \times 10^8$, with "Ra" defined using the characteristic length, $L = A/p$, where "A" is the surface area and "p" is the perimeter.

Cichelli and Bonilla (20) reported on experimental work conducted to determine the superheat temperature difference required to cause boiling from a horizontal chromium-plated surface at pressures ranging from atmospheric to near critical, for different fluids such as water, ethanol, benzene, propane, n-pentane, n-heptane, and different mixtures of these fluids. They found that the coefficient of heat transfer continues to rise with pressure until nucleate boiling ceases to be stable near the thermodynamic critical pressure, with the boiling surface becoming vapour-bound in most cases.

An approximate graphical correlation is presented for predicting the maximum boiling rate possible at any pressure, and another for the temperature difference required to produce the maximum rate. This maximum rate is found to occur at about one-third of the critical pressure. Binary mixtures were found to require several times as large a temperature difference to boil at the same rate as the pure components. The curves and formulas reported, according to the authors, enable the estimation of nucleate boiling heat transfer coefficients from horizontal surfaces or tubes for any liquid under any conditions.

Evaporation

During the process of evaporation (or condensation), there is always a mass movement of vapour molecules from (or toward) the vapour-liquid interface. Schrage (21) considered the effect of this mass vapour movement and derived a correction factor for the calculation of the the rate of phase change. Maa (22) performed experiments designed to examine Schrage's method of computing the effect of mass vapour movement on the rate of phase change. The experimental apparatus used was the jet stream tensimeter. This paper provides a useful model for determining the rate of evaporation as a function of interfacial cross-sectional area, film and saturation temperatures, and interfacial heat transfer coefficients.

Bailey et al. (23) determined the liquid stratification in the feed system of liquid-fueled rockets. The stratification is caused by the high temperatures experienced and results in a subsequent loss in Net Positive Suction Head (NPSH) of pump-fed engines. This technique provides a means of determining the rate of change of volume in the stratified interface region. The stratification analysis is based on integration of liquid mass flow in a turbulent natural convection boundary layer along the heated tank wall. The boundary layer mass flow is assumed to collect in a warm upper stratified layer called the upper stratum. This will enable a simplified evaluation of the heat transfer coefficient required to determine the rate of evaporation in the present report.

Previous studies of natural convection in cavities only considered cavities of rectangular shape, vertical cylinders and completely-filled horizontal cylinders. Aydemir (24) studies the problem of transient natural convection in a partially-filled, uniformly-heated horizontal cylinder. Numerical methods are used to solve the problem and the results show similar general trends of those observed experimentally.

Flashing Bubble Rise and Void Fraction

Peterson et al. (26) investigated two types of phase change:surface evaporation and flashing. Both processes are initiated by a reduction in pressure at the liquid/vapour interface and results in the cooling of the liquid due to the removal of energy in the form of latent heat of vapourization. This paper provides experimental data for the heat and mass transfer rates encountered during sudden depressurization. The thermal behavior of a layer of Freon-11 subject to a sudden depressurization, which corresponds to imposing an initial superheat on the liquid, is studied by interferometry. Experiments were performed to examine the subsequent mass transfer rates encountered in flashing and evaporation. The experiments were performed with temperatures ranging from 12.8°C to 20.8°C and an imposed pressure drop of 40.6 kPa to 54.2 kPa. Over this range, a mean value of the ratio of mass rate due to flashing to mass rate due to evaporation is 11.6. The higher rate of mass transfer encountered during flashing are attributed to: i) severe agitation causing high rates of evaporation, ii) entrainment of drops of subcooled liquid with the departing vapour.

Harmathy (27) proposed a new method of calculating the terminal velocity of fluid particles moving in media of either infinite or restricted extent. The paper deals only with drops and bubbles which have a Reynolds number larger than 500 (when viscous effects become negligible). Relations to determine particle velocity as a function of bubble shape, density of the particles and media, surface tension and Eotvos number (Eo), where $Eo = g\Delta\rho d^2/\sigma$, are presented. In this expression, $\Delta\rho$ is the difference in densities between the fluid particle and the media. The bubble shapes considered are oblate ellipsoidal, spherical cup, and cylindrical. The effect of wall proximity is also investigated. It is found that in turbulent conditions the drag coefficient is independent of Reynolds number and the fluid particle shape is solely responsible for variations in the drag coefficient. The relationship presented correlates well with experimental data except for liquid particles falling in vapour media.

Gaertner et al. (28) used electroplating as a means to determine the number of nucleation sites throughout most of the nucleate region for a boiling liquid. The technique for determining the active sites consisted of plating a thin layer of nickel on a cooper surface during the boiling runs and subsequently counting the number of pin holes in the plate. The heat flux was varied from 7680 to 535,000 $Btu/hr/ft^2$ and the superheat from 17.3 to 218.8°F. Counts of active sites were obtained from zero to a maximum of 1130 m^2. The results indicate the linear relationship between heat flux and the number of nucleation sites suggested by Jacob (29) does not apply. Instead, the heat flux is approximately proportional to the square root of the number of nucleation sites. The experiments are only conducted at atmospheric conditions; therefore no information is available to relate the number of nucleation sites to the pressure. These authors, however, note that in Kurihari and Myers' work (30), it is concluded that surface roughness and the nature of the fluid have no effect on the heat transfer coefficient and the active site population.

A paper by Wilson et al. (31) reported on an experimental program to determine the magnitude of the steam void fraction, during the process of bubbling steam through saturated water in a pressure vessel. The experiment is conducted to determine the "real" water level which may differ from the "gravitation" water level. The results are correlated for the void fraction as a function of density, surface tension, geometry, and bubble velocity. All tests and correlations are conducted for water only. The extension of these correlations for other fluids, such as propane, will require experimental validation.

Tong (32) evaluated the departure velocity of bubble rise in a gravitational field. Since the flow path is very short, the departure velocity may be used to approximate the bulk bubble velocity. The resulting correlation has not dependence on void fraction. This information and information from (31) enables the determination the void fraction of the tank car contents.

Friedel and Purps (33) reported on experiments performed on a vertical axis pressure vessel containing Freon-12. Systematic vessel depressurization is carried out to determine boiling delay, the two-phase level swell and the transient distribution of the mean void fraction across the vessel height. The effects of varying decisive parameters such as initial liquid level, relief cross section, starting pressure, and initial degassing are also examined. This work indicates that simplifying assumptions of constant void fraction or linearly ascending vapour concentration in vessels often used in computational algorithms may not apply.

EXISTING MODELS OF TANK CARS IN A FIRE ENVIRONMENT

A report by Graves (35) described a computer model for a tank car subjected to a fire environment. This model reduces the tank geometry to a cylinder of finite length and may accommodate various safety or relief valves. The model also has the capability of simulating the thermal response when the tank car is covered with up to two layers of insulation. The thermal conductivity of the insulation materials may vary with temperature and beyond some specified temperature, characteristic of each material, it is assumed that the material will start to deteriorate or flow.

The external wall boundary conditions may be imposed in the form of temperatures or heat fluxes, both of which may vary circumferentially and axially. However, the axial gradients must be small enough to be neglected since no provision was made for axial conduction through the shell. The tank fluid is allowed to coexist in the liquid and vapour phases, with the liquid always saturated and the vapour optionally either saturated or superheated. The temperature for each phase is considered uniform, but the two temperatures are not necessarily equal.

The time-dependent and space variable boundary conditions, the lading initial conditions, thermodynamic properties, equations of state and the specific heat equation for superheated vapour, all of which are used to calculate the heat transfer to the liquid and vapour, are supplied to the model. Simple equations for one-phase choked fluid flow through the valves are derived and used in the model. In addition, correlations of subcooled and moderate boiling and vapour space heat transfer coefficients are necessary.

The solution algorithm is iterative in nature and the calculation proceeds as outlined. For each time step, the total fluid mass is known and a pressure value throughout the fluid is assumed. Considering that both the liquid and the vapour phases are saturated, their specific volumes and internal phases are saturated, their specific volumes and internal energies are known, thermal equilibrium is assumed and the heat input is checked against the total internal energy. If their value exceeds some arbitrary tolerance value, a new guess for pressure is made and the calculation step is reinitiated. When the vapour phase is superheated, the equations for mass conservation and energy have to be solved simultaneously as the vapour temperature and specific volume are unknown.

Birk (36) considered each phenomenlogical process in great detail; however, simplifications are used to approximate the "real" behavior of the system. For instance, the model neglects the free convection and conduction in the liquid during the initial stages and instead uses a heat transfer coefficient for boiling which will tend to increase the pressure response in the period prior to venting. This boiling coefficient is also considered to be constant throughout the whole process which does not correspond to reality. The linearized radiant heat transfer coefficient is calculated using the maximum wall temperature, resulting in a large rate of heat transfer to the vapour space. For the two-phase frothing region, a constant heat transfer rate has been assumed and the height of the frothing layer is considered constant. The equation of state of the tank lading is approximated using Starling's equations of state (37).

EXPERIMENTAL RESULTS FROM FIELD TESTS

Anderson et al. (38) reported on the first full-scale fire tests conducted as part of a research program jointly sponsored by FRA/DOT and the RPI-AAR Tank Car Safety Project. The test car was an uninsulated 33,600-gallon tank car filled with 133,000 lbm (31,900 gallons) of LPG. It was positioned upright on its trucks in an excavated pit. A JP-4-fueled fire was maintained at a constant level in the pit. The tank car was equipped with a spring-loaded safety relief valve set to open at approximately 280 psig. The objectives of the tests were to gather data on tank car failure modes, the fire environment and the response of the car for use in the development of a theoretical model.

The tank ruptured after 24.5 minutes and at this time, it was estimated to be half full. The skin temperature in the region of the initial fracture site was of the order of

1200°F. The maximum pressure reached was 350 psig, but the pressure had dropped to 335 psig at the time of rupture.

A large number of thermocouples were located in the lading, on the interior wall and in the fire. Other instrumentation included pressure gauges, a radiometer, calorimeter and liquid level gauges. The mean heat flux of the fire at a point four feet about the pool was 43,920 BTU/hr/ft^2 but 90 percent of the time the heat flux was below 64,080 BTU/hr/ft^2 and above 35,280 BUT/hr/ft^2. Fire temperatures ranged from 1200°F to 2000°F. Temperatures at the east end of the tank were generally 360°F lower than those at the west end. The temperature of the propane torch from the relief valve was estimated at 2000°F, although the instrumentation failed early in the test.

Calculations were made on the heat input to be wetted surface area of the tank and the mass flow rate venting through the relief valve. These calculations required knowledge of the liquid level in the tank at certain times. The three liquid level measuring devices installed in the tank failed before the test, or early in the test, and no meaningful data were obtained. An indirect method of determining liquid level was then employed using the interior wall thermocouple readings. These readings rose suddenly as the liquid level receded past the thermocouples. The mass flow rate of the valve was calculated to be 73 lbm/sec with a discharge coefficient of 0.86. This was over a pressure range of 320 to 350 psig.

An expression was also derived to obtain the heat flux to the wetted surface. This was determined over time intervals when the pressure was relatively constant. The time weighted mean was determined to be 33,230 BTU/hr/ft^2.

Early in the test when fire temperatures were at their highest levels, the vapour pressure in the tank rose rapidly to about 270 psig at which point the safety relief valve opened. After cycling three times, it remained open throughout the test and was functioning when the tank ruptured.

The destructive testing to be used for comparison in this report is described in (39). The series of tests were performed to determine the behavior of protected and unprotected tanks and in particular, the effectiveness of an expanded aluminum foil honeycomb-like structure (EXPLOSAFE), used to reduce the hazard of Boiling Liquid Expanding Vapour Explosion (BLEVE). The testing was comprised of subjecting six one-fifth linear scale vessels laden with propane to fire conditions similar to those occurring in accidents.

The specific test used for comparison of this computer model is test #6-0000-4 which was performed on the vessel labeled ENIGMA. The vessel was 3.353 m in length, 0.6096 m in diameter with a shell thickness of 6.4 mm. The vessel was also fitted with a pressure relief valve with a rated capacity of 56,067 litres/min of air at 2.07 x 10^6 N/m^2.

The instrumentation on the vessel was required to provide information on circumferential wall temperatures, vapour space and liquid space temperature distribution, radial temperature profiles in the vertical, oblique and horizontal planes, flame temperature distribution and internal pressure.

A grid of thermocouples was designed to fulfill these requirements with a total of 43 thermocouples inside and outside the tank. The spacing of the radial thermocouples was chosen to yield equal temperature increments on the assumption that the temperature profiles would be logarithmic. Additional grid points provide more complete information with regard to liquid level. The wall thermocouples were equally spaced at 22.5 degree intervals (i.e., a circumferential distance of 117.2 mm).

All thermocouples were 1.5 mm in diameter, stainless-steel sheathed, closed junction "J" type for internal locations and "K" type for external locations. The wall thermocouples were welded to the inside wall and potted in Sauereisen cement. Grid thermocouples were attached to a cruciform-like structure attached to the vessel by tackwelds and located 50.8 mm from the measurement plane. Two pressure taps were located at the top of the vessel.

Since the test did not incorporate a pool fire calorimeter, the only means of rating the fire intensities is to determine the rate of heat input into the vessels through an energy balance. This has been computed for each vessel on an incremental basis using three different methods, all involving an energy balance and the change in thermodynamic state of the lading. From the report no correlation is apparent between the measured mean fire temperature over the wetted surface area and the heat flux. This points out the potential inaccuracies of attempting to describe and to analyze the pool fires on the basis of temperature measurement.

MATHEMATICAL MODEL

Physical Processes

The heat transfer occurring between the fire engulfing the tank and its lading is a complex process. The modes of heat transfer that can be identified are as follows: i) radiation from the flames plus convection from the combustion gases and heated air to the external walls of the tank; ii) radial and circumferential conduction and thermal storage in the tank walls; iii) heat transfer to the lading liquid and vapour phases evolving through the following modes: conduction/free convection, subcooled boiling, saturated nucleate pool boiling and possibly film pool boiling accompanied by a dramatic escalation of tank wall temperatures. The increase in bulk temperature results in a corresponding increase in pressure eventually causing the pressure relief valve to open. As tank levels drop due to boil-off and venting, heat transfer above the liquid between the tank wall and vapour will be by free convection and radiation, with some additional forced convection when venting through the pressure relief valve occurs.

Model Description

The model of the tank car considers the propane lading to be subdivided into two thermodynamic control volumes. The top region is assumed to contain vapour propane which may be either superheated or two-phase and the bottom region contains liquid propane which may be either subcooled or two-phase. With these possible states, there are only four possible conditions which may exist within the railway tank car shell as shown in Table 1.

TABLE 1

Possible States Within the Tank

State Designation	Description of State Within Tank
1	Top superheated, bottom subcooled
2	Top superheated, bottom boiling
3	Top condensing, bottom subcooled
4	Top condensing, bottom boiling

Each condition is shown in Figure 1 with the possible mass and energy flows at each condition. Equations are derived for each of the possible conditions which may occur in the tank car.

Assumptions

In order to develop a model for such a complex system, some simplifying assumptions are required. The assumptions, based on some experimental evidence (25, 39) used in this model derivation are as follows:

i) the only modes of energy and mass transfer across the interface are due to evaporation and bubble rise;

ii) bubble rise only occurs when the lower control volume is in two-phase state (i.e., saturated conditions prevail);

iii) bubbles leaving the liquid region do so with the vapour saturation enthalpy corresponding to the vessel pressure;

iv) the bubbles reach terminal velocity instantaneously;

v) Harmathy's correlation (27) for terminal bubble rise velocity is applicable to this geometry;

vi) the latent heat of the mass leaving the bottom control volume through evaporation is that corresponding to the vessel pressure;

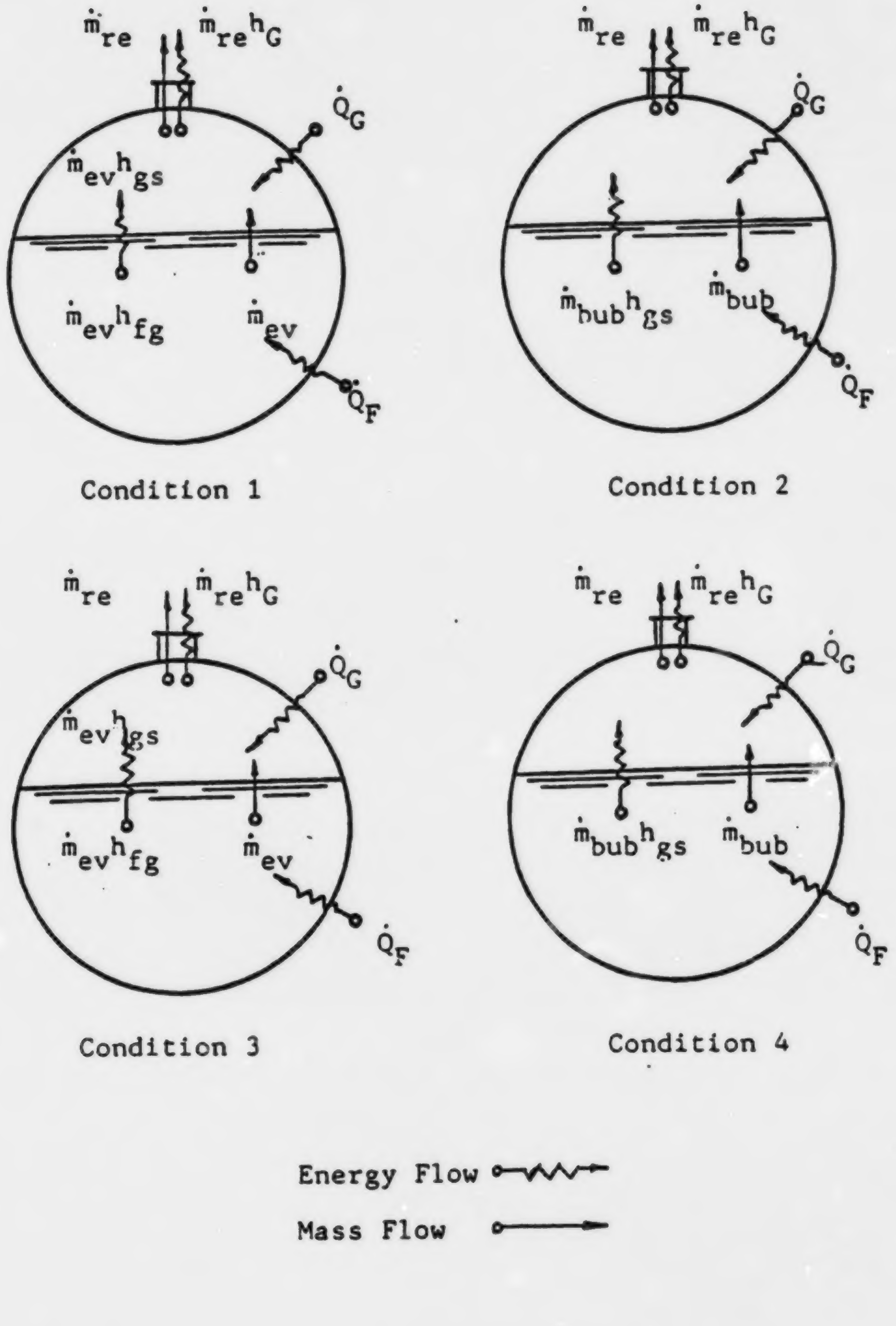
$\dot{m}_{re}$
$\dot{m}_{re}h_G$
$\dot{Q}_G$
$\dot{m}_{ev}h_{gs}$
$\dot{m}_{ev}h_{fg}$
$\dot{m}_{ev}$
$\dot{Q}_F$
Condition 1
$\dot{m}_{re}$
$\dot{m}_{re}h_G$
$\dot{Q}_G$
$\dot{m}_{bub}h_{gs}$
$\dot{m}_{bub}$
$\dot{Q}_F$
Condition 2
$\dot{m}_{re}$
$\dot{m}_{re}h_G$
$\dot{Q}_G$
$\dot{m}_{ev}h_{gs}$
$\dot{m}_{ev}h_{fg}$
$\dot{m}_{ev}$
$\dot{Q}_F$
Condition 3
$\dot{m}_{re}$
$\dot{m}_{re}h_G$
$\dot{Q}_G$
$\dot{m}_{bub}h_{gs}$
$\dot{m}_{bub}$
$\dot{Q}_F$
Condition 4
Energy Flow
Mass Flow

Figure 1. Mass and energy fluxes for each condition.

vii) mass entering the top control volume through evaporation does so with the saturated vapour enthalpy corresponding to the vessel pressure;

viii) condensation on the internal tank shell and in the bulk vapour region is neglected;

ix) over a time step, uniform values for properties and parameters are considered throughout each control volume;

x) pressure is uniform throughout the tank car over each time step;

xi) the mass flow through the pressure relief valve is considered to follow a polytrophic process;

xii) the radiative component of heat flux input to the exterior of the tank car shell proposed in (16) is used;

xiii) the tank car shell is homogeneous with constant density, specific heat and conductivity; and

xiv) the volume of the tank car remains constant.

Governing Equations

Conservation equations of mass, energy and volume and equations of state for the two control volumes are coupled together in order to obtain an explicit equation for the time derivative of pressure (dP/dt). Once pressure is determined the other time-dependent derivatives of properties can be determined; thus, given initial conditions at time "t", the new conditions at time "t+dt" can be determined. The governing equations for each possible condition are unique to each condition and are presented in the following pages.

State 1: Top superheated and bottom subcooled

Equation of state: $v_F = f_F(P,h_F)$; $v_G = v_G(P,h_G)$

$$XB = 0.0 \tag{1.1}$$

$$XT = 1.0 \tag{1.2}$$

$$V_F = m_F v_F \tag{1.3}$$

$$V_G = m_G v_G \tag{1.4}$$

$$\dot{m}_F = -\dot{m}_{ev} \tag{1.5}$$

$$\dot{m}_G = -\dot{m}_{re} + \dot{m}_{ev} \tag{1. 6}$$

$$(mh)_{Fi} = \dot{Q}_F - \dot{m}_{ev} h_{fg} \tag{1.7}$$

$$(mh)_{Gi} = \dot{Q}_G - m_{re} h_G + m_{ev} h_{gs} \tag{1.8}$$

$$\begin{aligned} \dot{P} = \{&\dot{m}_F v_F + \dot{m}_G v_G + \partial v_F/\partial h_F\{(mh)_{Fi} - \dot{m}_F h_F\} + \partial v_G/\partial h_G\{ \\ &(mh)_{Gi} - \dot{m}_G h_G\}\}/\{m_F \partial v_F/\partial P + m_G \partial v_G/\partial P + V_F \partial v_F/ \\ &\partial h_F + V_G \partial v_G/\partial h_G\} \end{aligned} \tag{1.9}$$

$$\dot{h}_F = (mh)_{Fi} - m_F h_F + PV_F/m_F \tag{1.10}$$

$$\dot{h}_G = (mh)_{Gi} - m_G h_G + PV_G/m_G \tag{1.11}$$

$$\dot{m}_{Ff} = \dot{m}_F \tag{1.12}$$

$$\dot{m}_{Fg} = 0.0 \tag{1.13}$$

$$\dot{m}_{Gf} = 0.0 \tag{1.14}$$

$$\dot{m}_{Gg} = \dot{m}_G \tag{1.15}$$

State 2: Top superheated and bottom two-phase

Equation of state:

$$v_{fs} = v_{fs}(P);\quad v_{gs} = v_{gs}(P);\quad v_G = v_G(P,h_G)$$

$$h_{fs} = h_{fs}(P);\quad h_{gs} = h_{gs}(P)$$

$$XB = m_{Fg}/m_F \qquad (2.1)$$

$$XT = 1.0 \qquad (2.2)$$

$$V_F = m_{Ff}v_{fs} + m_{Fg}v_{gs} \qquad (2.3)$$

$$V_G = m_Gv_G \qquad (2.4)$$

$$\dot{m}_F = -\dot{m}_{bub} \qquad (2.5)$$

$$\dot{m}_G = \dot{m}_{bub} - \dot{m}_{re} \qquad (2.6)$$

$$(mh)_{Fi} = \dot{Q}_F - \dot{m}_{bub}h_{gs} \qquad (2.7)$$

$$(mh)_{Gi} = \dot{G}_G + \dot{m}_{bub}h_{gs} - \dot{m}_{re}h_G \qquad (2.8)$$

$$MFH = m_{Ff}\partial h_{fs}/\partial P + m_{Fg}\partial h_{gs}/\partial P-V_F$$

$$MFV = m_{Fg}\partial v_{gs}/\partial P + m_{Ff}\partial v_f/\partial P$$

$$\begin{aligned}\dot{P} = -\{&h_{fg}\{\dot{m}_Gv_G + \dot{m}_Fv_{gs} + \partial v_G/\partial h_G\{(mh)_{Gi} - \dot{m}_Gh_G\}\}\\ &+ v_{fg}\{(mh)_{Fi} - \dot{m}_Fh_{gs}\}\}/\{h_{fg}\{MFV + m_G\partial v_G/\partial P\\ &+ V_G\ \partial_G/\partial h_G\} - MFHv_{fg}\}\end{aligned} \qquad (2.9)$$

$$\dot{h}_F = 0.0 \qquad (2.10)$$

$$\dot{h}_G = \{(mh)_{Gi} - \dot{m}_Gh_G + \dot{P}V_G\}/m_G \qquad (2.11)$$

$$\dot{m}_{Ff} = -\{(mh)_{Fi} - \dot{m}_Fh_{gs} - MFH*P\}/h_{fg} \qquad (2.12)$$

$$\dot{m}_{Fg} = \dot{m}_F - \dot{m}_{Ff} \qquad (2.13)$$

$$\dot{m}_{Gf} = 0.0 \qquad (2.14)$$

$$\dot{m}_{Gg} = \dot{m}_G \qquad (2.15)$$

State 3: Top two-phase and bottom subcooled

Equation of state:

$v_{fs} = v_{fs}(P); \quad v_{gs} = v_{gs}(P); \quad v_F = v_F(P, h_F)$

$h_{fs} = h_{fs}(P); \quad h_{gs} = h_{gs}(P)$

$$XB = 0.0 \tag{3.1}$$

$$XT = m_{Gg}/m_G \tag{3.2}$$

$$V_F = m_F v_F \tag{3.3}$$

$$V_G = m_{Gf} v_{fs} + m_{Gg} v_{gs} \tag{3.4}$$

$$\dot{m}_F = -\dot{m}_{ev} \tag{3.5}$$

$$\dot{m}_G = -\dot{m}_{re} + \dot{m}_{ev} \tag{3.6}$$

$$(mh)_{Fi} = \dot{Q}_F - \dot{m}_{ev} h_{fg} \tag{3.7}$$

$$(mh)_{Gi} = \dot{Q}_G - \dot{m}_{re} h_G + \dot{m}_{ev} h_{gs} \tag{3.8}$$

$$MGH = m_{Gf}\,\partial h_{fs}/\partial P + m_{Gg}\,\partial h_{gs}/\partial P - V_G$$

$$MGV = m_{Gg}\,\partial v_{gs}/\partial P + m_{Gf}\,\partial v_{fs}/\partial P$$

$$\begin{aligned} P = {} & -\{v_{fg}\{(mh)_{Gi} - \dot{m}_G h_{gs}\} + h_{fg}\{\dot{m}_G v_{gs} + \dot{m}_F v_F + \{(mh)_{Fi} \\ & - \dot{m}_F h_F\}\,\partial v_F/\partial h_F\}\}/\{h_{fg}\{MGV + m_F\,\partial v_F/\partial P + V_F\,\partial v_F/\partial h_F\} \\ & - v_{fg} MGH\} \end{aligned} \tag{3.9}$$

$$\dot{h}_F = \{(mh)_{Fi} - \dot{m}_F h_F + \dot{P} V_F\}/m_F \tag{3.10}$$

$$\dot{h}_G = 0.0 \tag{3.11}$$

$$\dot{m}_{Ff} = \dot{m}_F \tag{3.12}$$

$$m_{Fg} = 0.0 \tag{3.13}$$

$$\dot{m}_{Gf} = -\{(mh)_{Gi} - \dot{m}_G h_{gs} - MGH{*}\dot{P}\}/h_{fg} \tag{3.14}$$

$$\dot{m}_{Gg} = \dot{m}_G - \dot{m}_{Gf} \tag{3.15}$$

State 4: Top two-phase and bottom two-phase

Equations of state:

$$v_{fs} = v_{fs}(P); \quad v_{gs} = v_{gs}(P); \quad h_{fs} = h_{fs}(P); \quad h_{gs} = h_{gs}(P)$$

$$XB = m_{Fg}/m_F \tag{4.1}$$

$$XT = m_{Gg}/m_G \tag{4.2}$$

$$V_F = m_{Ff}v_{fs} + m_{Fg}v_{gs} \tag{4.3}$$

$$V_G = m_{Gf}v_{fs} + m_{Gg}v_{gs} \tag{4.4}$$

$$\dot{m}_F = -\dot{m}_{bub} \tag{4.5}$$

$$\dot{m}_G = \dot{m}_{bub} - \dot{m}_{re} \tag{4.6}$$

$$(mh)_{Fi} = \dot{Q}_F - \dot{m}_{bub}h_{gs} \tag{4.7}$$

$$(mh)_{Gi} = \dot{Q}_G + \dot{m}_{bub}h_{gs} - \dot{m}_{re}h_G \tag{4.8}$$

$$MFH = m_{Fg}\,\partial h_{gs}/\partial P + m_{Ff}\,\partial h_{fs}/\partial P - V_F$$

$$MFV = m_{Fg}\,\partial v_{gs}/\partial P + m_{Ff}\,\partial v_{fs}/\partial P$$

$$MGH = m_{Gg}\,\partial h_{gs}/\partial P + m_{Gf}\,\partial h_{fs}/\partial P - V_G$$

$$MGV = m_{Gg}\,\partial v_{gs}/\partial P + m_{Gf}\,\partial v_{fs}/\partial P$$

$$\begin{aligned}\dot{P} = {} & -\{v_{fg}\{(mh)_{Gi} + (mh)_{Fi} - (\dot{m}_G = m_F)h_{gs}\} \\ & +h_{fg}\{v_{gs}(\dot{m}_G + m_F)\}\}/\{h_{fg}(MGV+MFV) - v_{fg}(MGH \\ & + MFH)\}\end{aligned} \tag{4.9}$$

$$\dot{h}_F = 0.0 \tag{4.10}$$

$$\dot{h}_G = 0.0 \tag{4.11}$$

$$\dot{m}_{Ff} = -\{(mh)_{Fi} - \dot{m}_F h_{gs} - MFH*\dot{P}\}/h_{fg} \tag{4.12}$$

$$\dot{m}_{Fg} = \dot{m}_F - \dot{m}_{Ff} \tag{4.13}$$

$$\dot{m}_{Gf} = -\{(mh)_{Gi} - \dot{m}_G h_{gs} - MGH*P\{/h_{fg} \tag{4.14}$$

$$\dot{m}_{Gg} = \dot{m}_G - \dot{m}_{Gf} \tag{4.15}$$

Determination of State

The pressure and enthalpy are updated at the end of each time step and these updated properties serve as the initial conditions for the next time step. The saturated values of enthalpy corresponding to the updated value of pressure are then determined. If the updated value of enthalpy is less than the saturated liquid enthalpy then the state of the fluid is subcooled. If the updated value of enthalpy is greater than the saturated vapour enthalpy then the state of the fluid is superheated. If the updated value of enthalpy lies between the saturated liquid and vapour enthalpies then the state of the fluid is two-phase.

Phenomenlogical Relations

To determine the net mass and energy flows into each of the control volumes, phenomenological relations are required. There is always heat input to both regions from heat conducted through the tank car shell. When the liquid region is subcooled, mass and energy is transferred to the vapour region through evaporation. When the liquid region is two-phase, mass and energy is transferred to the vapour region in the form of bubble rise. When the tank pressure increases to the point where it is sufficient to activate the pressure relief valve there is both a mass and an energy transfer through the pressure relief valve. These energy and mass transfers are discussed in more detail in the following section. The determination of the equations of state are also briefly discussed in this section.

Flame model

The flame model is used to evaluate the heat flux imposed on the tank car external surfaces. The heat flux incident on the tank car is due to radiation from the flame, and convection from the combustion gases and heated air.

Temperature histories for the fire surrounding both the vapour space and the liquid space are presented in (39). These temperatures appear to vary linearly with time from ambient temperature to a maximum fire temperature at approximately 60 seconds whereupon the fire temperature remains relatively constant. Since these temperatures were measured with thermocouples, which are relatively insensitive to radiation, the true value of temperature may differ from that measured but the general trend of the temperature history is assumed to agree with the measured temperatures.

There are three classifications of fires presented in (15): catastrophic, uncontrolled and controlled. The type of fire used in the experimental one-fifth scaled models (39) are catastrophic in which a fully enveloping fire incurs a heat flux on the tank car with little wind effects. In (15), it is reported that a catastrophic fire involving propane can reach heat fluxes up to 108.8 kW/m^2 on a full-scale tank car. This value for the maximum heat flux will be used as a constraint in this simulation model.

Radiation

Birk and Oosthuizen (16) have produced several numerical results of radiant heat flux on a horizontal cylindrical surface as a function of flame temperature, flame absorption coefficient, cylinder and surrounding temperatures, and circumferential position. Several plots of heat flux as a function of circumferential position for differing controlling parameters are presented. One of the major assumptions in this flame model is that the flame temperature and the flame absorption coefficient remain constant throughout the process.

One of the presented plots of radiant flame heat flux has characteristics very similar to the field tests, that being:

1) Flame temperature = 870°C
2) Flame absorption coefficient = 0.656/m
3) Cylinder wall and surrounding temperatures = 15.5°C

A third order polynomial has been fitted to this graph as a function of circumferential position yielding.

$$q/A_{rad\ max} = 70.1784 - 12.4548\theta - 4.97278\theta^2 + 1.35473\theta^3$$

Assuming that the emissivity remains constant throughout the heating process the following relationship for radiative heat transfer is used:

$$q/A_{rad} = q/A_{rad\ max} * (T_{fire}^4 - T_{wall}^4)/(1143^4 - 288.5^4)$$

Although Tunc and Venart (17) have performed a more detailed model of the radiant heat flux to the tank car than in (16), the results presented in (17) do not have similar characteristics to the actual fire conditions, therefore the radiation model presented in (16) will be used.

Convection

A total heat flux (THFM) of 108.8 kW/m^2 (15) will be assumed to occur when the fire temperature is 1143K and the tank car external surface is at 288.5K up to 90 degrees along the circumference where it is assumed to linearly decrease to the radiative component at the top of the tank car shell (Figure 2). Since the temperature difference between the fire and tank car surface is never this great, the standard maximum heat flux of 108.8 kW/m^2 associated with catastrophic fires should be not be exceeded.

Since the total heat flux on the external surface of the tank car is made up of radiation and convection, the amount that the total heat flux exceeds the radiative component is assumed to be due to convection. Considering a constant convective heat transfer coefficient, the heat flux due to convection is a linear function of the fire to wall temperature difference. This yields the following relation for convection heat transfer incident on the tank car surfaces:

$$q/A_{conc} = (THFM - q/A_{rad\ max})*(T_{fire} - T_{wall})/(1143-288.5)$$

Internal Heat Transfer Coefficients

The internal heat flux is determined by the fluid temperature, the internal surface shell temperature, and the heat transfer coefficients.

Vapour Space Heat Transfer Coefficients

The vapour space heat transfer is initiated by a short period of conduction which, in turn, sets up convective currents causing convection to be the major mode of heat transfer. Radiation becomes an important component of heat

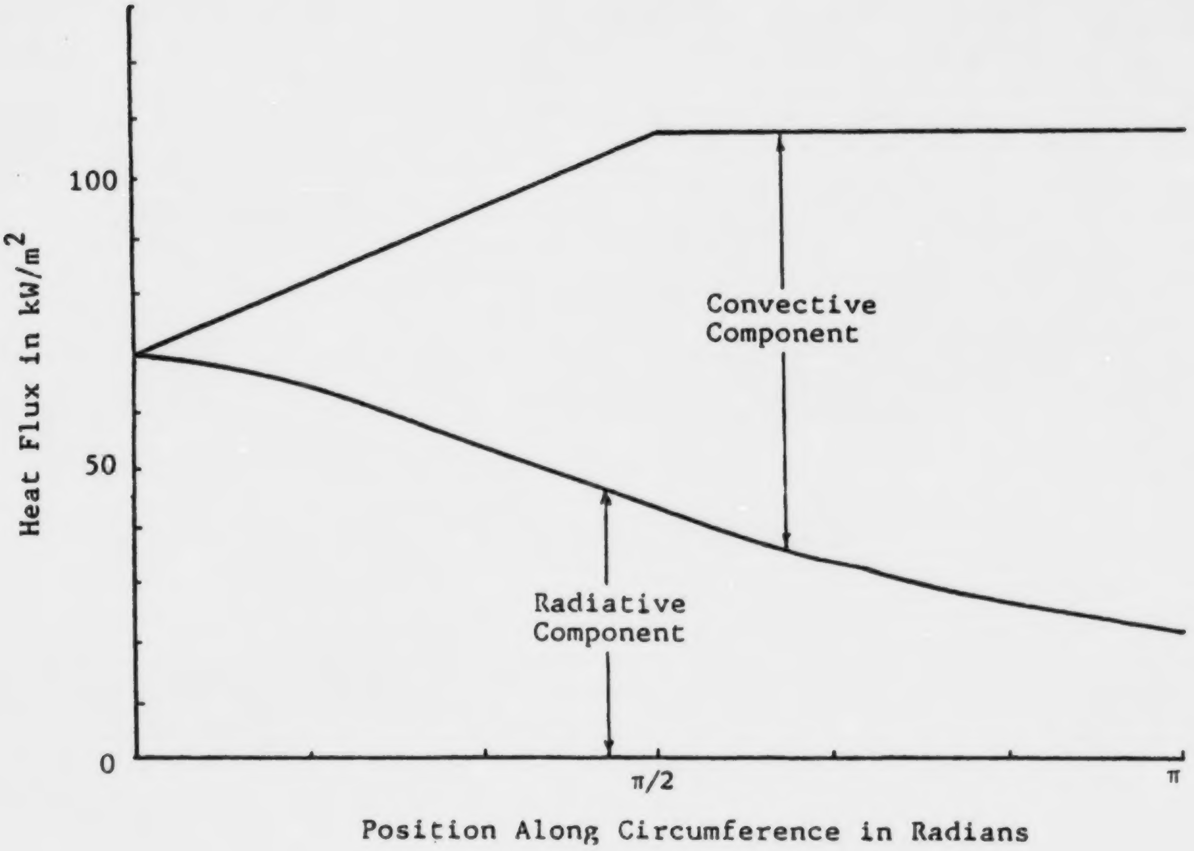

Figure 2. Heat flux imposed by enveloping fire with T_{fire} = 1143K and T_{wall} = 288.5K

transfer as the absolute temperature and the wall to fluid temperature difference increases.

The calculation of heat transfer to the vapour space is simplified by neglecting the initial period of conduction and the component of heat transfer due to radiation. This leaves convection as the only mode of heat transfer to the vapour space. Neglecting the conduction component should have little effect on the results since it occurs for only a short period; however, the exclusion of the radiation component will have a more pronounced effect on the results, in particular on the shell temperatures. If the radiation component were included, there would be more heat transfer from the shell thus reducing the shell temperature. However, since the vapour, except for high absorption bands, may be considered relatively transparent to radiation, neglecting the radiation term should have little effect on vapour temperature.

Convection

The shell surface bounding the vapour space will be assumed to be a flat plate which is inclined at some angle. For the period of interest (prior to venting), the liquid level remains relatively constant. Therefore the flat plate will be assumed to have the length of the chord between the top of the internal surface of the shell and the top of the liquid surface at the angle which it encompasses with the vertical (refer to Figure 3). With a known initial fill level both   and L can be determined from the geometry.

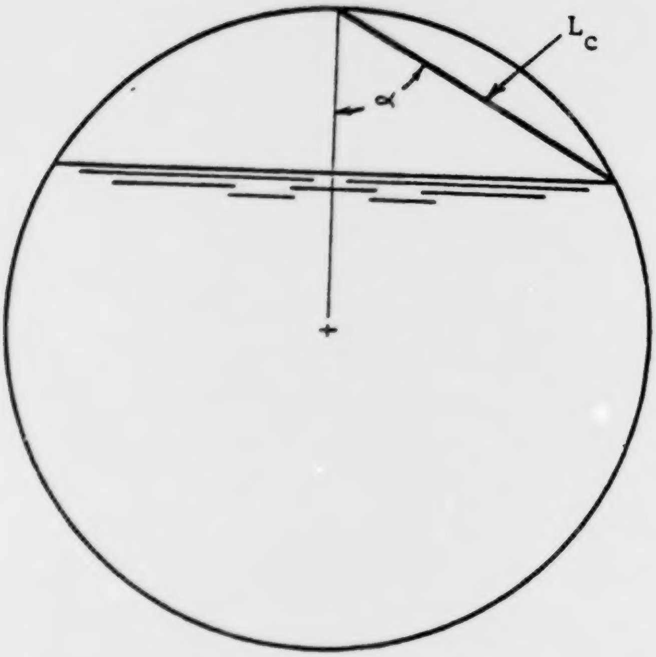

Figure 3. Vapour space heat transfer coefficient model.

Fujii and Imura (41) using water as the experimental fluid, presented a relationship for the heat transfer coefficient for inclined surface as

$$Nu = 0.56\ (Gr\ Pr\ \cos\alpha)^{1/4}$$

for : $\alpha < 88$ degrees

$$10^5 < Gr\ Pr\ \cos\alpha < 10^{11}$$

where the Nusselt number (Nu)

$$Nu = h\ L_c/k$$

All the properties in Nu except β are evaluated at temperature T_e

where $T_e = T_w - 0.25\ (T_w - T_\infty)$

is evaluated at the temperature T

where $T = T_\infty + 0.25\ (Tw - T_\infty)$.

Also $Gr\ PR = Ra = g\ \beta\ (Tw - T_\infty) L_c{}^3 c_p/(\mu v^2 k)$

In applying these relations over the expected temperatures and pressure ranges expected prior to venting, it was discovered that the resulting vapour heat transfer coefficient did not vary significantly, therefore a constant value, reducing the computation effort, should introduce little error. Using the mean temperatures and pressures over the period prior to the pressure relief valve opening and assuming perfect gas behavior, results in a heat transfer coefficient of 22 W/m^2K. This value has been verified through the experimental results of the one-fifth scaled tests as well.

Vliet and Ross (42) presented a correlation for the heat transfer coefficient from inclined surfaces using data produced with air as the experimental fluid. Again, the mean properties and parameters prior to relief valve opening were used in the correlation. The heat transfer coefficient determined from this method was approximately 27 W/m^2K. Since this value does not differ significantly from the heat transfer coefficient determined from the method described in (41), the value of 22 W/m^2K will be used in this simulation.

Liquid Space Heat Transfer Coefficients

Over a short initial period, conduction prevails producing thermal gradients which, in turn, initiates free convection. This free convection may initially be in the laminar region; however, as the wall-to-liquid temperature difference increases, turbulent free convection becomes the predominant mode of heat transfer. As the film temperature near the wall exceeds the propane saturation temperature corresponding to the internal pressure by a few degrees, localized nucleate boiling (or subcooled boiling) will result. Once the bulk fluid reaches saturation temperature at the corresponding internal pressure, pool boiling will be the predominant node of heat transfer. Each of these modes of heat transfer will be discussed and models will be presented.

Natural convection

Assuming each shell element to be a hot isothermal vertical plane, McAdams' correlation (43) and Warner's correlation (44)

$$Nu = 0.59\ Ra^{1/4} \text{ for } 10^4 < Ra < 10^9$$

$$Nu = 0.1\ Ra^{1/3} \text{ for } 10^9 < Ra < 10^{13}$$

respectively, may be applied, where the Nusselt Number (Nu) is

$$Nu = h\ L_C/k$$

and the Rayleigh Number (Ra) is

$$Ra = g\ \beta(T_w - T_\infty)L_C{}^3 cp/(\mu v^2 k)$$

The characteristic length for a vertical plane is the height of the plane, so the circumferential distance from the bottom of the tank to the top of the liquid surface will be used. A constant value of the characteristic length is used to reduce the computational complexity in calculating a liquid space free convective heat transfer coefficient. This is in keeping with the accuracy of the problem since free convection should not prevail beyond for the first 30 seconds into a run (25).

Chapman (47) gave the following correlation as an approximate rule for evaluating the coefficient of thermal expansion of liquids,

$$\beta = 0.06284\ (T_C - T_\infty)^{-0.641}$$

where "T_C" is the critical temperature of the liquid in degrees Rankine. It should be noted that this empirical rule may become inadequate at extreme temperatures but it is useful when other data are lacking.

Nucleate boiling

For the case of nucleate boiling, Change and Synder (45) proposed the following pool nucleate boiling Reynolds number

$$\frac{L^{*}G^{*}}{\mu} = \frac{F(p)}{[q/A/(\Delta T\ c_p\ G^{*})]^{a}[P_r]^{b}}$$

where:

$$L^{*} = \frac{c_p\ (\rho_F - \rho_{gs})T_s\,\sigma}{(\rho_{gs} h_{fg}^{2})\ J} \qquad G^{*} = \frac{(q/A)\rho_F}{h_{fg}\ \rho_{gs}}$$

$$F(p)=57.2*10^{-4} \qquad a=2.4 \qquad b=1.0$$

Rearranging the above correlation with h = q/(AΔT) yields:

$$h = \frac{(F(p)J)^{1/2.4} c_p (q/A)}{h_{fg}} \left(\frac{v_{gs}}{v_{fs}}\right) \left[\frac{c_p^{2}\sigma T_s\ (q/A)}{k\ (h_{fg})^{3}} \left(\frac{v_{gs}}{v_{fs}}\right)^{2} (v_{gs} - v_{fs})\right]^{-1/2.4}$$

This correlation will be considered valid for both saturated boiling and subcooled boiling. This should introduce little error if the added resistance to bubble generation due to the subcooled core can be neglected.

Jung et al. (46) have considered the effects of the angle of inclination and the heat flux on the free convective, nucleate and film boiling regions. Freon -11 is used as the experimental fluid at pressure ranging from 1 to 2 bars. This work is not considered relevant to the present study since the pressure range is much different than that encountered in the tank car and the fluid is different than that carried in tank cars.

Shell Heat Transfer

Unsteady conduction is considered due to the continually changing boundary conditions on the internal and external tank car surface. The axial gradient of heat conduction is considered to be negligible; therefore, only a two-dimensional analysis is required. Due to the geometry of the tank car shell, the most convenient choice for the coordinate system is polar coordinates. The differential two dimensional conduction equation to be solved is of the form:

$$\rho c \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r}\left(rk \frac{\partial T}{\partial r}\right) + \frac{1}{r} \frac{\partial}{\partial \theta}\left(\frac{k}{r} \frac{\partial T}{\partial \theta}\right)$$

The methodology used for solving the heat conduction through the shell is described in (49). The two dimensional differential equation for conduction through the shell is discretized using the control volume formulation. The final discretized equation is of the form:

$$a_P T_P = a_E T_E + a_W T_W + a_N T_N + a_S T_S + B$$

where a_E, a_W, a_N, a_S = interface conduction terms

a_{PO} = term associated with heat storage

a_P = $a_E + a_W + a_N + a_S + a_{PO}$ in the system

T_E, T_W, T_N, T_S = new surrounding nodal temperatures

T_P = new temperature of node under consideration

B = source term

The source term is made up of terms associated with the previous time step and the boundary conditions (the heat input term and the convection term). Unless the node in consideration is located on either the internal or external surface, the heat input term and the convection are zero for that node. If the node in consideration is on the internal surface, the convection term will not be zero, and if the node in consideration is on the external surface, the heat input term will not be zero. The Crank-Nicolson scheme is used to increment the solution in time in which the temperature distribution is equally dependent on the temperature distribution from the previous time step and the new time step; therefore, the solution is iterative in nature for each time step.

The boundary conditions for the solution of this equation vary with time due to the nature of the problem. The inside boundary condition is dependent on a heat transfer coefficient and the fluid temperature of the previous time step. The heat transfer model takes into account the different modes of heat transfer as previous discussed. The outside boundary condition will be specified by the heat flux distribution of the fire, based on the model described previously. Also, due to symmetry, only one half of the tank needs to be considered.

Evaporation

Evaporation is the only mode of mass and energy exchange across the interface when the liquid control volume is subcooled. The evaporation model consists of a boundary layer which travels along the vessel wall and collects to form a stratified layer at the liquid/vapour interface (upper stratum). Some of this upper stratum enters the vapour region through evaporation and some remains in the liquid control volume. An explanation of this model is described in more detail in this section.

Bailey et al. (23) presented a relationship to determine the rate of change of volume in the upper stratum as:

$$\dot{V}_H = 4.0\ q\ A\ /\ [c_p\ \rho\ (T_w - T_F)]$$

where "q" is the heat flux to the liquid region and "A" is the wetted area of the inner surface of the tank shell. So, the product of "q A" is the heat input rate to the liquid region. Values for specific heat and density are evaluated at the upper stratum conditions. A mean value of the temperature of the wall in contact with liquid will be used for T_W and T_F is the liquid bulk temperature.

Bailey et al. (23) assumed that the boundary layer is in the turbulent regime. This should not present a problem since the experimental results indicate that turbulence begins very early in the tests. This boundary layer moves along the wall and is collected in the upper stratum.

Once the rate of change of volume in the upper stratum is determined from equation (16), the mean volume over that time step is multiplied by the current time step value in order to determine the amount of volume flow into the stratum region. The film thickness, , is then approximated by dividing the total volume flow over the time step by the cross-sectional area of the interface.

In order to determine the amount of the upper stratum which leaves the liquid region, an evaporation model is required. Maa (22) presented the following relationship for the mass rate of evaporation assuming that the latent heat is removed from the liquid through conduction

$$k(\partial T/\partial x) = h_{ev}(T_{st} - T_{fs}) = h_{ev}\dot{m}_{ev}$$

The interfacial heat transfer coefficient, h_{ev}, must be evaluated to determine the mass flow rate, m_{ev}. In order to solve for "h", a linear interfacial temperature gradient across the upper stratum is assumed. Upon discretizing, the following result is obtained:

$$(\partial T/\partial x) = (T_{st} - T_{fs})/\delta$$

This substituted into equation (17) yields:

$$h_{ev} = k/\delta$$

With the interfacial heat transfer coefficient calculated, it is possible to evaluate the mass flow rate due to evaporation and its associated energy flux from equation (17).

Bubble Rise

A bubble rise model independent of geometry is desired in order to make the overall model as general as possible. The bubble rise model which is used in this project is similar to that presented in (5) and its general outline is as follows:

$$\dot{m}_{bub} = m_F X_B V_t / L_B$$

where: m_{bub} = mass flow rate of the bubble rise kg/sec

m^F = total mass in the liquid region in kg

X_B = bubble mass fraction

V_t = terminal velocity of the bubbles in m/sec

L_B = mean distance the bubble must travel to reach the interface, in metres

The terminal velocity of the bubbles rising is determined from Harmathy's correlation (27) for terminal bubble velocity given by:

$$V_t = \{0.0153(g(\rho_{fs} - \rho_{gs})\partial/\rho^{2}_{fs}{}^{2})\}^{0.25}$$

where: V_t = terminal velocity of bubbles rising in m/s

g = acceleration due to gravity (981 cm/sec)

ρ_{fs} = density of saturated liquid in g/cc

ρ_{gs} = density of saturated vapour in g/cc

∂ = surface tension in dynes/cm (=g/sec^2)

The mass fraction of bubbles to total liquid is approximately as the quality of the two-phase propane in the lower region, which has been determined from the governing equations.

Mass Flow Rate Through the Pressure Relief Valve

This model only considers the case when the tank car is upright. Consequently, the only region to see the pressure relief valve is the vapour region. The flow model assumes single phase flow following a polytropic process. Knowing that the pressure set of the pressure relief valve exceeds the pressure required for choked flow, the following relationship for mass flow is used:

$$m_{re} = C_d A_{ef} \rho_t \sqrt{\gamma R T_t}$$

where: m_{re} = mass flow rate in kg/sec

C_d = discharge coefficient

A_{ef} = effective valve flow area in m^2

γ = ratio of specific heats: c_p/c_v

T_t = temperature at the throat in K

R = gas constant for propane

ρ_t = density of the vapour

Temperature at the throat, effective valve throat area and the density of the vapour are the only variables in this equation. The valve will have a high set value and a low set value. The valve is considered to be fully open at the high setting and the effective area decreases linearly until the pressure reaches the low setting where it closes completely. The temperature and density at the throat are determined assuming an isentropic expansion through the pressure relief valve.

Since the initial liquid level is quite high, liquid entrainment may occur during which a discharge coefficient greater than 1.0 is required. In this model, a constant discharge coefficient of 0.86 is used, as reported in (38).

Equations of State

Since the equations of state for propane are not readily available over the range required for the model, another approach must be taken to evaluate the propane properties and partial derivatives. The approach taken in this model was to generate thermodynamic tables for propane from a computer program presented in (34). The program evaluates various properties of propane as a function of temperature and density. From these inputs, a number of flags are used to determine what state the fluid is in, and then the computer generates the associated properties. If the propane is in the two-phase region, the saturated properties corresponding to the input temperature are evaluated. If the propane is in a single-phase region, both density and temperature are required to determine the corresponding properties. Many temperatures and densities were used to generate the thermodynamic tables of propane over the required range. Although these tables are only valid up to 700°K, they have been extended in the superheated region by assuming a perfect gas behavior.

Once these tables were generated, the input arguments for the single-phase regions had to be changed to pressure and enthalpy, and for the two-phase region, to pressure. This is performed in the single-phase regions by using a double interpolation technique using second order Lagrange polynomials for the fitting function. In the two-phase region, the only input required is pressure and again second order polynomials are used for the interpolation.

The interpolation tables are limited to a maximum pressure of 30.0 bar and a maximum temperature of 1200K. The lower limits of the interpolations are a pressure of 2.0 bar and a temperature of 240K. This range of properties easily fulfills the requirements of the simulation.

METHOD OF COMPUTATION

The calculation procedure for each condition is shown in chronological order in the section "Governing Equations". In each condition, the first calculation determines the quality of each region, based on updated values of the masses, m_F, m_{Fg}, m_G, m_{Gg}. This calculation procedure is not required when the region in consideration is single phase. If the top region is superheated, XT = 1.0, if the bottom region is subcooled, XB = 0.0

Once the state of each region is determined, the program calls the appropriate subroutine to determine the other properties and the property derivatives as a function of pressure and enthalpy at the new time step. It is then possible to determine the volume each region occupies based on the updated masses and the newly determined specific volumes, equations (i.3) to (i.4),, with i = 1, 2 , 3 or 4.

The auxiliary equations are used to determine the various mass flow rates and heat input into each region. The total mass and energy flux rates can be calculated as the sum of these rates as can be seen in equations i.5 to i.8.

Using equation, i.9, the derivative of pressure with respect to time can be determined explicitly from the information on mass and energy fluxes and the partial derivatives of properties. This also leads to the explicit calculation of $\dot{h}_F$ and $\dot{h}_G$.

The time derivatives of the mass terms can then be determined from the relations i.12 to i.15.

The values of enthalpies, pressure and masses are then updated through integration and the whole procedure is repeated again for each time step. There is an exception to the method of updating when the fluid is in the two-phase state. In this case, $\dot{h}_F$ or $\dot{h}_G$ is set equal to zero and the new value of enthalpy is calculated based on the newly updated values of quality and the saturated values of enthalpy at the updated pressure.

If a change of state is indicated after the updating procedure, the time step is repeated using the previous initial conditions and a new time step which is half of the original. This procedure is repeated until the change of state occurs at the minimum time step, which is set in the program geometrical statements. This is done because each set of equations in only valid for a specific condition within the tank car, and there is an inherent discontinuity with a change in condition. Therefore, the only way to reduce the error as the condition changes is to reduce the time step associated with the condition change.

RESULTS AND COMPARISON

The numerical simulation is validated for the case of an unprotected tank filled with propane. The results of the simulation are compared with experimental data collected during the one-fifth linear scale model test of an uninsulated tank (39). The specific test used for comparison was designated ENIGMA by the experimenting group.

The first simulation results to be compared are those with the modeling parameters described previously in the body of this report. Some of these controlling parameters are arbitrarily altered to test the performance of the model and to determine whether each of the varied parameters are significant in the model. These parametric analyses are designated as follows:

1. Modeled results
2. Vary the vapour space heat transfer coefficient
3. Vary the heat input
4. Vary the vapour generation

In the evaluation of the model, attention is concentrated on the pre-venting response of the tank although model predictions and field results are presented for up to and including the second vent. The results to be compared are: internal pressure, bulk liquid temperature, bulk vapour temperature, and internal liquid and vapour space shell temperature.

Case 1: Modeled Results

Figure 4 shows the variation of the internal tank pressure with respect to time. It can be observed that the model predicts with reasonable accuracy the internal pressure up to and including the first valve opening of the pressure relief valve. The model predicts a valve opening time of 142 seconds while the experimental results indicate a time of 138 seconds. The nature of the two curves are also very similar. They increase exponentially with time, and this is due to the fact that both heat input from the flame and the internal liquid space heat transfer coefficient increase with time. Although the valve simulation displays the ability to cycle, the pressure response during the cycling does not correlate well with the experimental results. This is due to the fact that valve characteristics are unique for each valve and if these characteristics are not thoroughly understood, an accurate model cannot be made.

Figure 5 was obtained by plotting the mean temperature of the liquid in the experimental vessel and the bulk temperature of the liquid computed from the model, as a function of time. Both the field results and the calculated results suggest that the bulk liquid is subcooled; however, the calculated results are slightly higher than those indicated in the field tests. One factor contributing to this effect is the fact that the experimental liquid temperatures are severely stratified and the calculation of the mean temperature may be in error. The method of calculating the mean temperature from experimental data only accounts for vertical stratification and not the radial temperature gradients near the walls, which increases with time and therefore would also increase the mean temperature if it were included in the calculation. However, since the correct profile of temperature near the wall vicinity is not known, an accurate calculation of mean liquid temperature cannot be performed. It is anticipated that an accurately calculated mean liquid temperature would improve the comparison between predictions and experimental data.

Figure 6 shows the variation of vapour temperature, experimental and calculated, as a function of time. The experimental values used are those indicated at the centre of the vapour space. These are considered to be more representative of the bulk temperatures than the mean values, again due to the uncertainty of the thermal stratification and since they correspond to quasi-stagnation points. There is good agreement between the experimental and calculated results although the calculated values of bulk vapour temperatures are slightly higher than the field values.

Figure 7 compares the internal liquid space shell temperatures at an angle of 135 degrees (the angle encompassed by the top of the shell, centre of the tank and the element position along the circumference of the tank). For the first 30 seconds, the agreement is only fair as the calculated values increase at a higher rate than the experimental values, but after the initial 30 seconds agreement is very good. This phenomenon can be explained by a change in the heat transfer regime within the model for the liquid heat transfer coefficient. The heat transfer coefficient is initially very low, causing a rapid increase in temperature, but as the heat transfer regime changes from free convection to boiling, the heat transfer rate increases substantially, causing a decrease in the rate of temperature increase in the shell. This process is somehow attenuated by the decreasing heat input from the fire to the tank with increasing shell temperature.

The vapour space internal surface temperatures indicated by the simulation correlate very well with the experimental results shown in Figure 8. The position of the element being compared is located at 22.5 degrees from the top of the shell along the circumference

Case 2: Vary the Vapour Space Heat Transfer Coefficient

Case 2 is similar to Case 1, except that the vapour heat transfer coefficient is varied from the modeled value of 22 W/m^{2}K. This parameter has been varied to determine the effect of the vapour space heat transfer coefficient on the thermal response of the tank vessel. Values of 10 to 40 W/m^{2}K have been chosen for the vapour space heat transfer coefficient.

Figures 4, 9 and 14 show that the pressure level response is slightly affected by a change in the vapour heat transfer coefficient. As the vapour space heat transfer coefficient is increase, so too does the pressure level of the tank vessel. The increase vapour space heat transfer coefficient substantially increases, the heat transfer to the vapour space resulting in a higher pressure level. This shows that although the vapour space heat transfer coefficient has only a slight effect on pressure, its effects cannot be neglected in pressure response evaluation.

The liquid space temperatures shown in Figure 5, 10 and 15 do not show any significant changes by varying the vapour space heat transfer coefficient.

There is a definite increase in bulk vapour space temperature with an increase in vapour space heat transfer coefficient as shown in Figures 6, 11 and 16. The changes in the vapour temperatures are quite drastic due to the vastly increased heat transfer to the vapour space accompanying the increased heat transfer coefficient.

The effects of varying the vapour space heat transfer coefficient on the liquid space shell temperatures are not really discernible from Figure 7, 12 and 17. Therefore, it can be concluded that the vapour heat transfer coefficient has little or no effect on liquid space shell temperature.

Figures 8, 12 and 18 show only a slight decrease in the vapour space internal surface shell temperatures, with an increase in the vapour space heat transfer coefficient. The decrease is only slight since the heat transfer rate to the vapour space shell greatly exceeds the heat transfer to the vapour space form the shell and the shell has a relatively large thermal storage as compared to the vapour space control volume

Case 3: Vary the External Heat Flux

In Case 3, all the controlling parameters remain the same as in Case 1, except the external heat flux from the flame is varied. The heat flux is varied by removing the convective component of heat flux (Figures 19-23), removing the radiative component of heat flux (Figures 24-28) and finally, by increasing the total maximum possible heat flux without changing the radiative component which essentially increases the convective component (Figures 29-33).

The plots used for comparing the tank pressure response are Figures 4, 19, 24 and 29. In Figure 19, it is obvious that the removal of the convective component reduces the level of pressure. The same situation occurs in Figure 24 with the removal of the radiative component. The pressure relief valve does not reach its opening pressure in the allotted time of 160 seconds in either case, but the removal of the convective term has a more pronounced effect. In comparing Figure 4 with Figure 29, the effect of changing the convective component is again realized. By increasing the maximum possible heat flux from 108 to 125 kW/m^2K, the time required for the pressure relief valve to activate is reduced from 142.0 to 129.5 seconds. The convective component having a more dramatic affect on pressure response is not really surprising on referring to the external heat flux model in the section "Flame Model". Figure 2 shows that the convective component of heat flux is much greater than the radiative component adjacent to the liquid space. Since the heat input to the liquid has a greater effect on pressure than the heat input to the vapour space, a change in the convective heat flux has a greater effect on pressure.

Figures 20, 25 and 30 show the liquid temperature response as the external heat flux is varied. From Figure 20, it can be seen that the liquid temperature is reduced considerably when the convective component of heat flux is removed, while Figure 25 shows that the liquid temperature does not display as significant a change with the removal of the radiative component of heat flux. This is due to the fact that the convection component is much greater than the radiative component on the shell adjacent the liquid space. In comparing Figure 30 with Figure 5, the liquid temperature responses are very similar although the increase in external heat flux slightly increases the liquid temperature level.

The results of the vapour space temperatures are plotted in Figures 21, 26 and 31. The only external heat flux component which as a great effect on the bulk vapour temperature in the radiative component. When it is removed (Figure 26), the bulk vapour temperatures predicted by the model are drastically underpredicted; however, when only the radiation component is present as in Figure 21, the bulk vapour temperature response is excellent as compared to the field results. When both components of heat flux are present with an increase in the convective component as in Figure 31, the model results overpredict the bulk vapour temperatures but it is obvious that the radiation component has a much greater effect on bulk vapour temperatures than the convective component. These results are not at all surprising when considering the heat flux model, as described previously. In Figure 2, it can be seen that the convective component of external heat flux adjacent to the vapour space is much less than the radiative component; therefore, the convection component has little effect on the results.

The liquid shell temperatures shown in Figures 22, 27 and 32 display similar trends to the bulk liquid temperatures previously discussed. The vapour shell temperatures are shown in Figures 23, 28 and 33. They display the same trends as the bulk vapour temperatures previously discussed. The reasons for the common trends are similar to those previously discussed and for this research they will not be repeated here.

Case 4: Vary the Vapour Generation

The controlling parameters of Case 4 are similar to those in Case 1, except that the vapour generation is varied. Figures 34-38 illustrate the results when the vapour generation is zero (i.e., no evaporation or bubble rise). Upon comparing Figures 34-38 with Figures 4-8, it is obvious that evaporation has very little effect on any of the results, which is not surprising considering the liquid core remains subcooled. This leads to the conclusion that evaporation effects are negligible on the tank car thermal response.

Figures 39-43 show the tank car thermal response when the vapour generation is increased by imposing bubble rise with a prescribed value of fluid quality of 0.0001. This results in a mean mass flow rate across the interface due to bubble rise of 0.018 kg/sec. The imposed bubble rise produces the expected results.

The pressure rise is much more dramatic than without any vapour generation as demonstrated in Figure 39. This effect occurs from a much increased rate of vapour generation displacing vapour volume in the vapour space.

The imposed bubble rise results in the liquid temperatures being slightly decreased (approximately $1^{\circ}C$ at 100 seconds) as can be seen by comparing Figures 5 and 40. The reason for the decrease in temperature is that the mass leaving the liquid space and entering the vapour space is of higher enthalpy than the enthalpy of the bulk of the liquid. This loss of energy is accompanied by a reduction of temperature in the liquid space.

The vapour temperature level is also decreased with the introduction of bubble rise. The reason for the temperature drop in the vapour space is that the mass entering the vapour space is of lower quality energy than the superheated vapour space mass. The low quality energy entering the vapour region reduces the energy level of the vapour space thus reducing the vapour space temperature. This effect can be seen upon comparing Figure 6 with Figure 41.

The liquid shell temperatures are slightly affected by the bubble rise mainly due to the decreased liquid temperatures. This effect is not really noticeable from Figures 7 and 42 but the temperature decrease at 100 seconds is approximately $1^{o}C$.

The vapour shell temperatures are not significantly changed when bubble rise is introduced as can be seen by comparing Figure 8 with Figure 43.

CONCLUDING REMARKS

The computer model developed to simulate the thermal response of LPG tanks engulfed in flames is benchmarked for pre-venting and initial venting conditions of one particular field test. Considering the random number of variables in this test, the predictions show an excellent agreement with the field data. The effects of varying some of the controlling parameters has also been investigated.

It was found that the vapour space heat transfer coefficient is an important factor in predicting bulk vapour space temperatures and when considering pressure response. However, heat input from the flame and vapour generation are clearly the dominating factors affecting the tank car thermal response. The effects of evaporation since the core is subcooled, were demonstrated to be insignificant; however, the imposed bubble rise affected the thermal response dramatically.

The fire itself is a complicated matter. The heat flux from a fire varies with the specific fuel, the geometry of the fire, degree of mixing of air and fuel, configuration of the radiating surfaces, emissivity of the flame, absorptivity of the tank surface, temperature variations in the flame and variations in the wind. In order to reduce the complexity of the model assumptions have been made in the flame model used in the simulation which have yet to be validated.

Bubble rise is also a very complicated subject. In the initial stages of the heating process, even prior to the valve opening, there is experimental evidence of subcooled boiling

(39). The subcooled boiling results in a mass transfer across the interface in the form of bubbles. It has already been demonstrated in Figures 39-43 that bubble rise is a very important factor in evaluating the tank car thermal response. If the mass flow due to bubble rise is of the same order of magnitude as that incurred through the imposed bubble rise, a significant variation in thermal response will result.

Even though the agreement between the field tests and the numerical predictions is good, one must be careful in attempting to evaluate the model. Before an accurate appraisal of the model can be drawn, external heat flux and bubble rise must be investigated more extensively. The test with which the model is compared provides very little information on these subjects. The tests were not conducted to collect experimental data for validating computer simulations but to study the ability of EXPLOSAFE material to reduce the hazard of BLEVEs when LPG tanks are exposed to fire.

RECOMMENDATIONS

Based on observations made during the numerical runs of the present computer program and from the comparisons of the predicted results with field test data it is recommended that:

1. Reliable experimental data be gathered in more field tests with smaller time steps between scans than used in previous tests.
2. More accurate measurements of flame properties and heat input to the tank should be evaluated to improve the model for the heat input to the tank car.
3. A model for vapour generation due to subcooled boiling prior to valve opening should be developed.
4. More thermocouples in the lading should be employed in any further destructive testing to better evaluate the liquid and vapour stratification.
5. The program should be further developed to simulate the tank car after the valve has been open with a more extensive evaluation of valve characteristics.

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Appendix I

COMPARISON PLOTS

This appendix presents plots of the computer simulation and the field results. The contents of the appendix and the associated figures are listed below.

1. Modelled results:	Figures 4-8
2. Vary the vapour space heat transfer coefficient (VHTC):	
i) VHTC=10 W/m^2K	Figures 9-13
ii) VHTC=40 W/m^2K	Figures 14-18
3. Vary the external heat flux:	
i) No convection heat flux component	Figures 19-23
ii) No radiation heat flux component	Figures 24-28
iii) THFM=125 W/m^2	Figures 29-33
4. Vary the vapour generation:	
i) No vapour generatoin	Figures 34-38
ii) Bubble rise with XB=0.0001	Figures 39-43

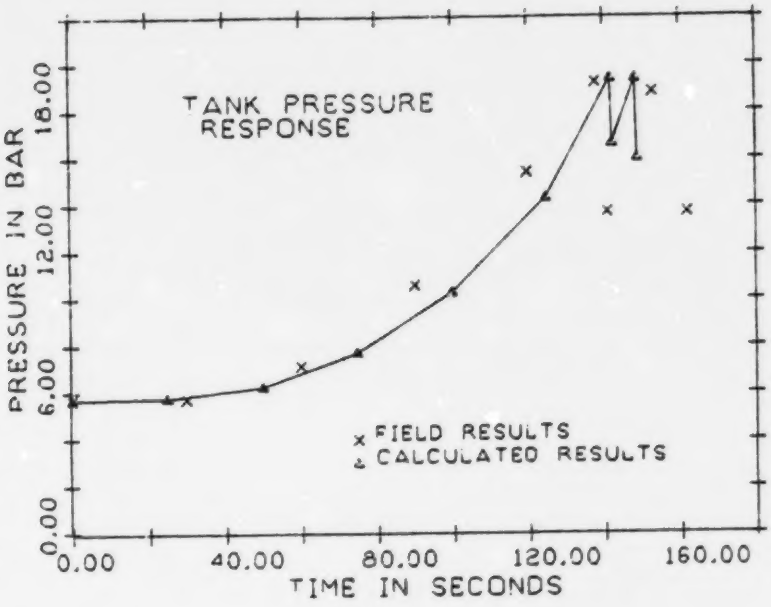


Figure 4. Pressure response comparison as modeled.

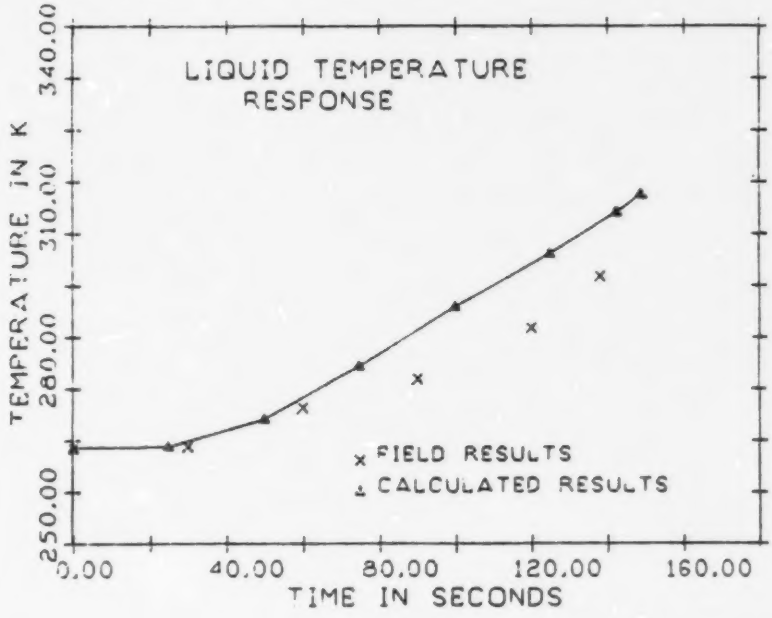


Figure 5. Bulk liquid temperature response as modeled.

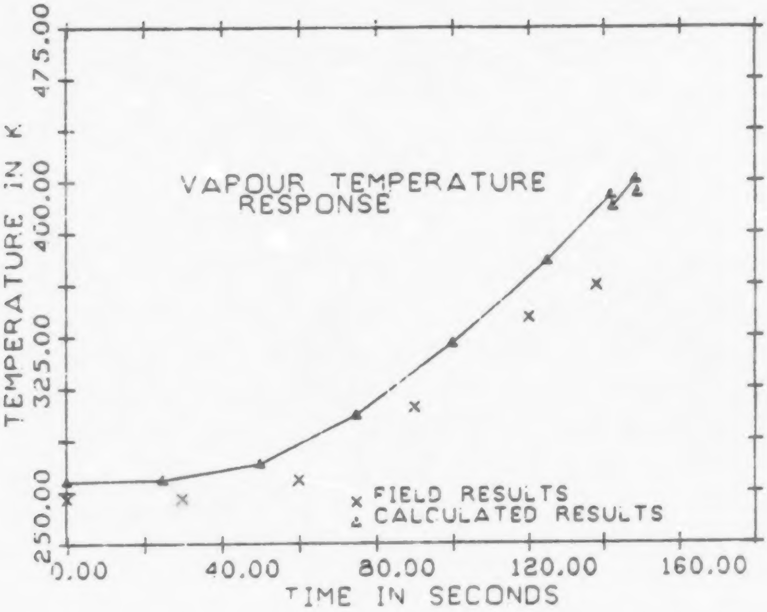


Figure 6. Bulk vapour temperature response as modeled.

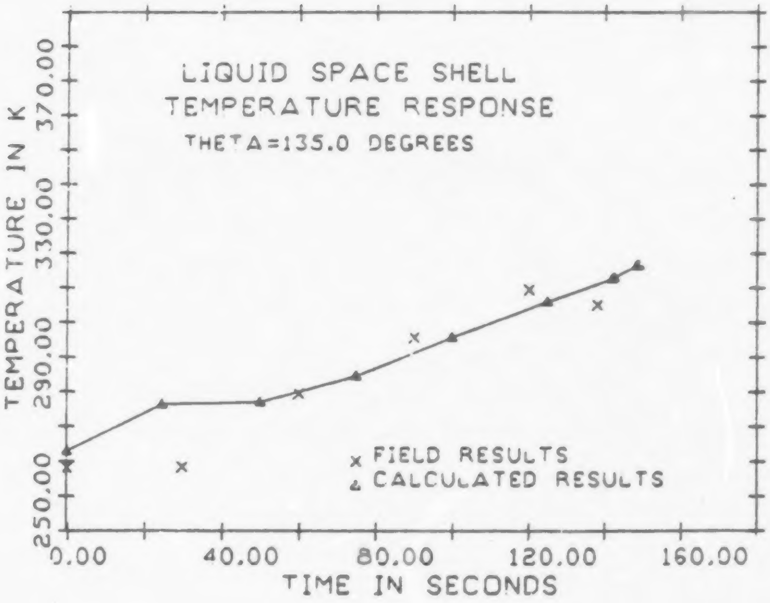


Figure 7. Liquid space shell temperature response as modeled.

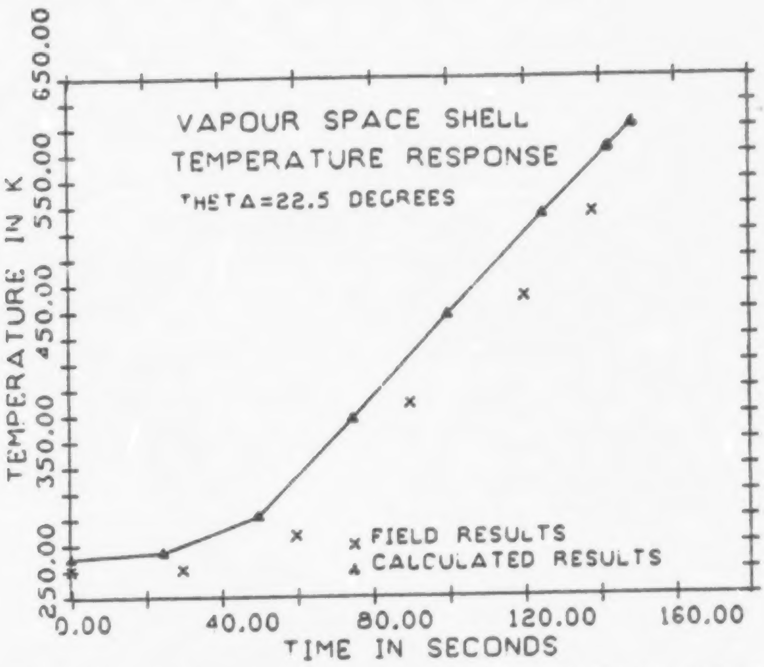


Figure 8. Vapour space shell temperature response as modeled.

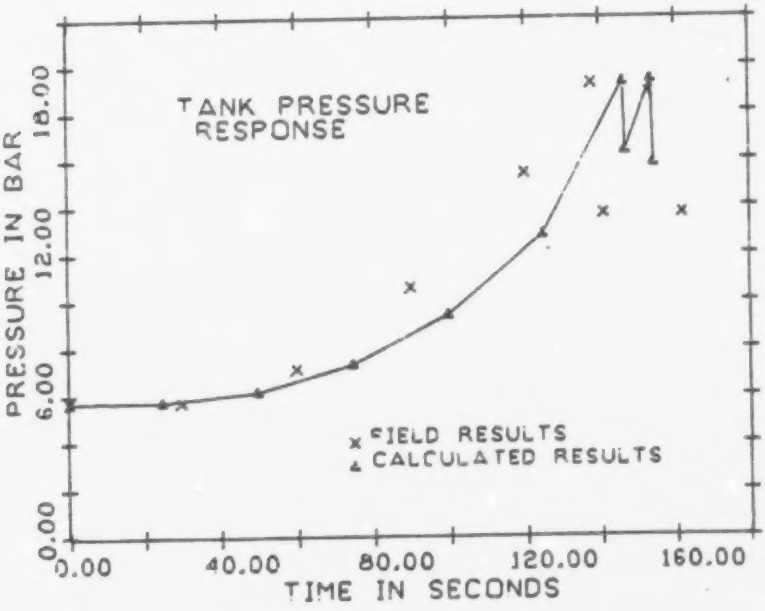


Figure 9. Internal pressure response with vapour space heat transfer coefficient of 10 W/K^2.

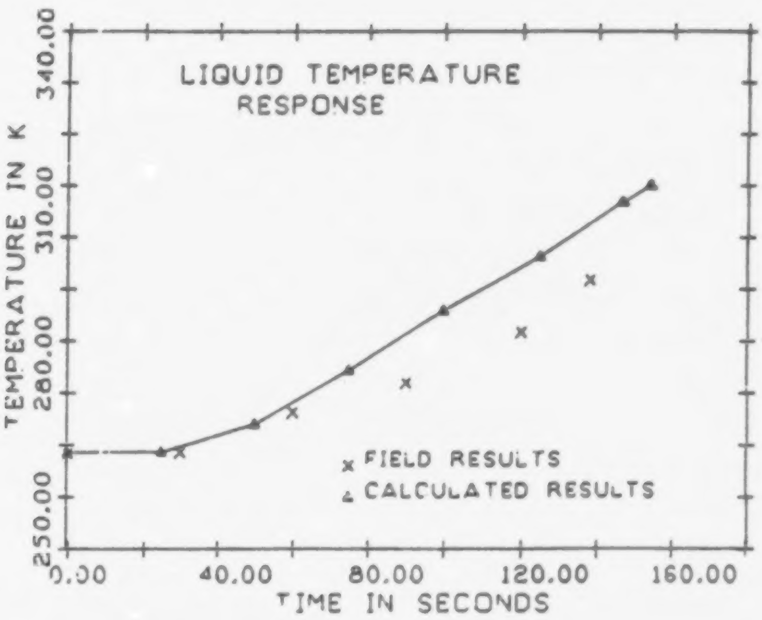


Figure 10. Bulk liquid temperature response with vapour space heat transfer coefficient of 10 W/K^2.

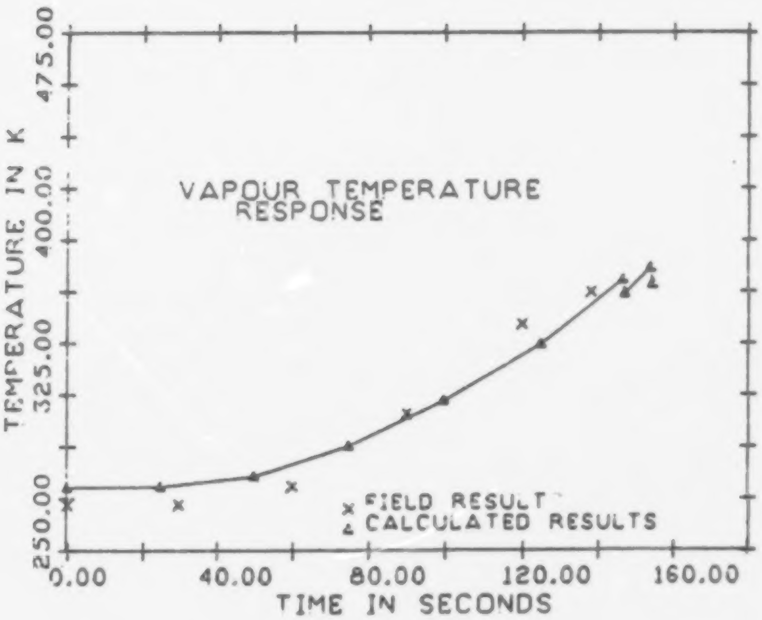


Figure 11. Bulk vapour temperature response with vapour space heat transfer coefficient of 10 W/K^2.

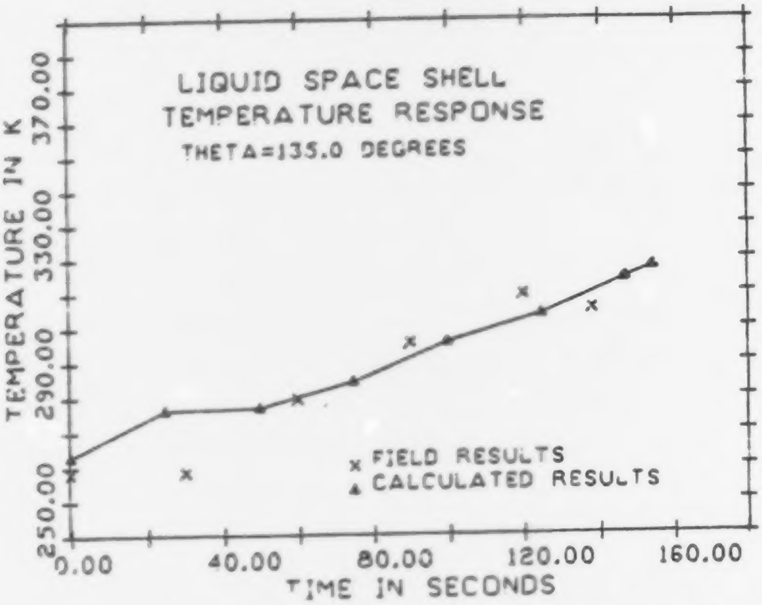


Figure 12. Liquid space shell temperature response with vapour space heat transfer coefficient of 10 W/K^2.

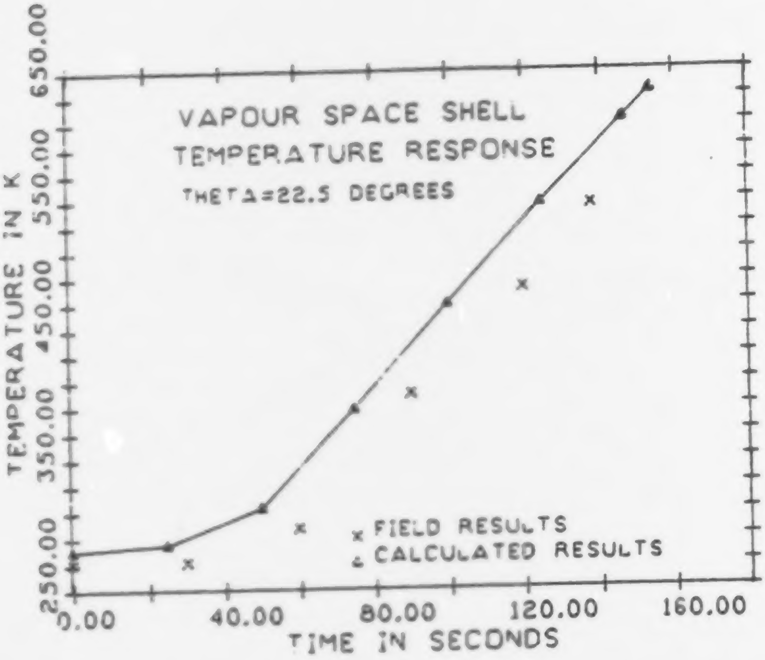


Figure 13. Vapour space shell temperature response with vapour space heat transfer coefficient of 10 W/K^2

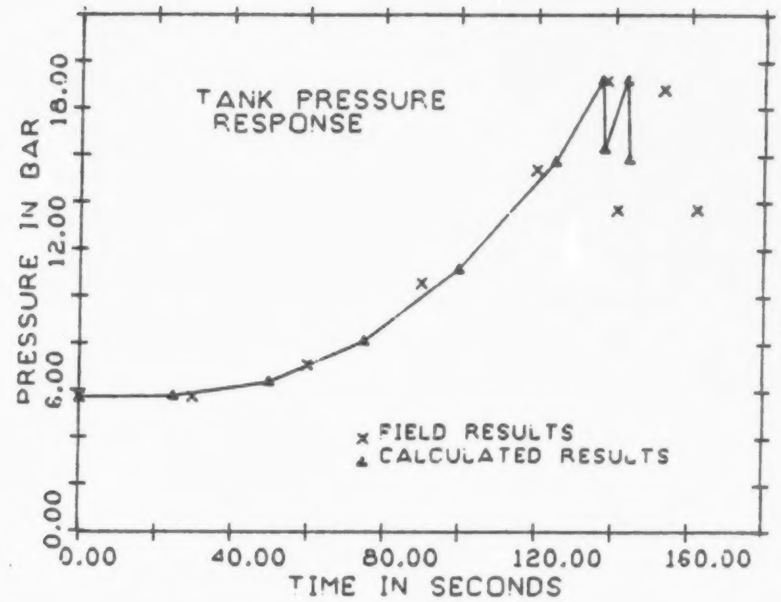


Figure 14. Internal pressure response with vapour space heat transfer coefficient of 40 W/K^2.

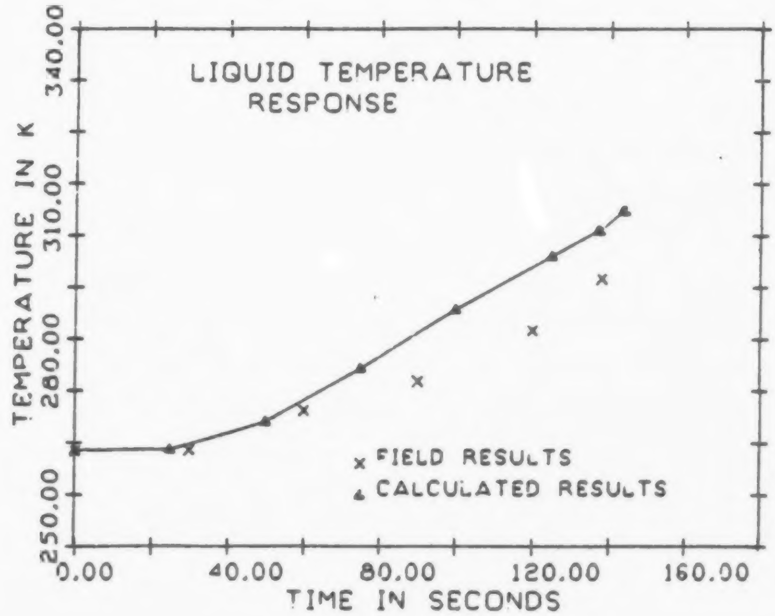


Figure 15. Bulk liquid temperature response with vapour space heat transfer coefficient of 40 W/K^2.

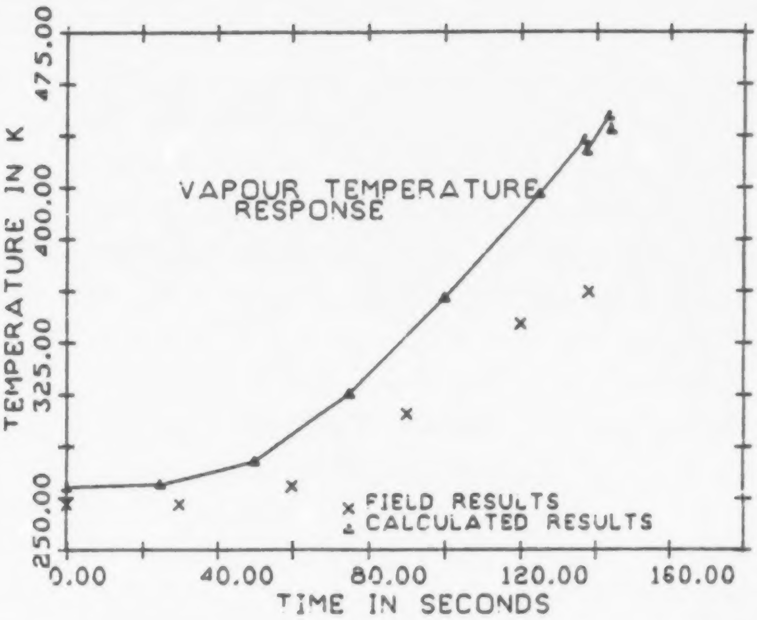


Figure 16. Bulk vapour temperature response with vapour space heat transfer coefficient of 40 W/K^2.

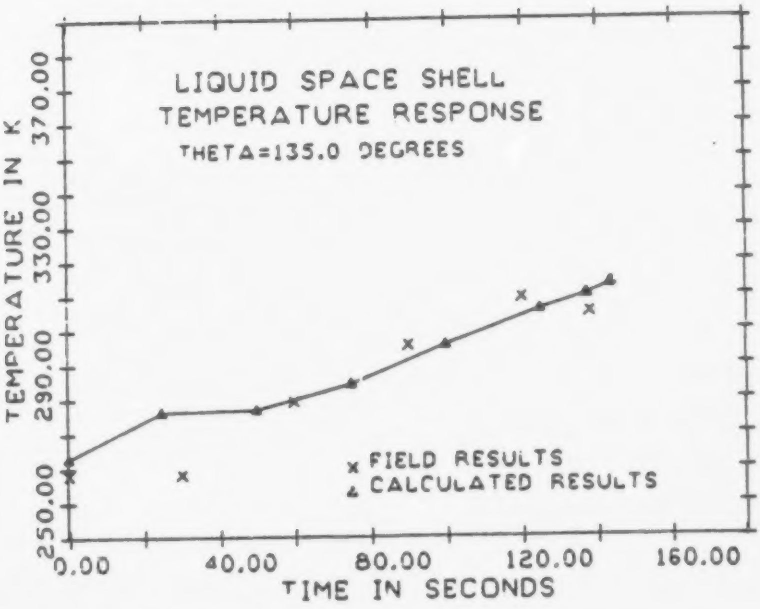


Figure 17. Liquid space shell temperature response with vapour space heat transfer coefficient of 40 W/K^2.

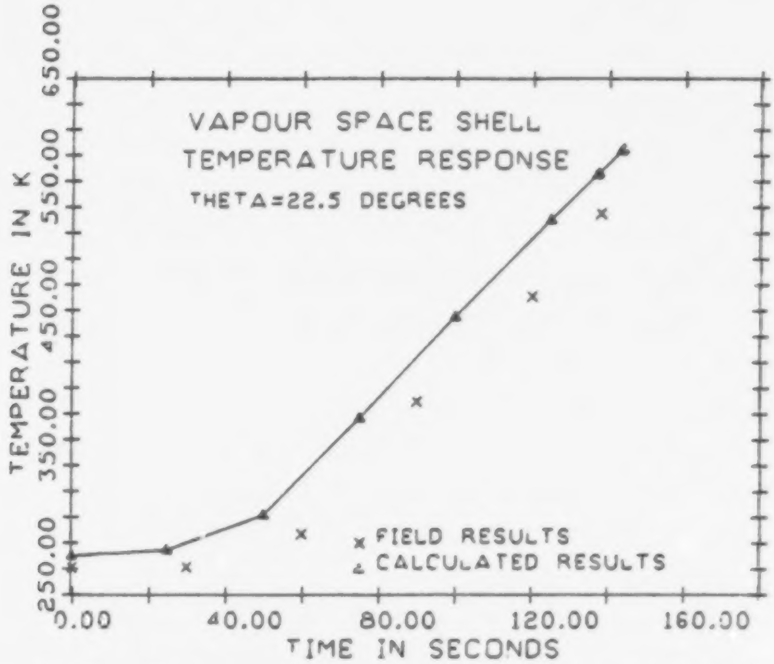


Figure 18. Vapour space shell temperature response with vapour space heat transfer coefficient of 40 W/K^2.

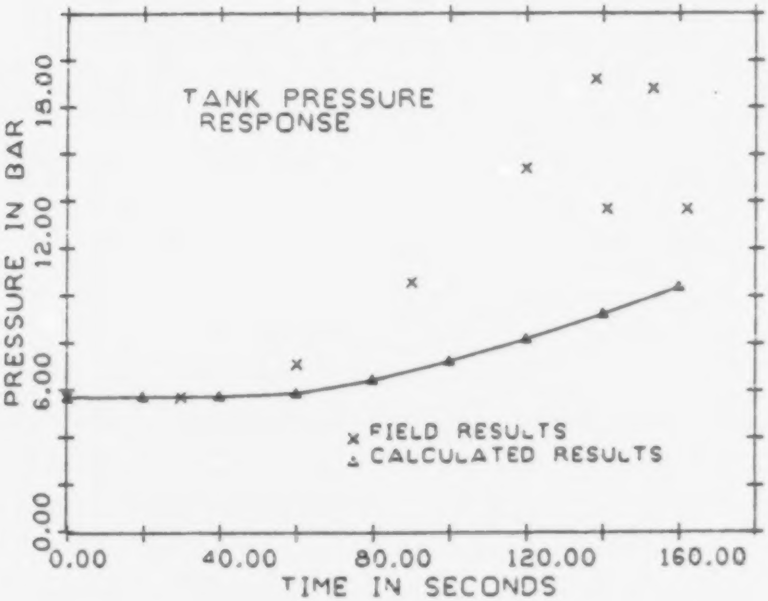


Figure 19. Internal pressure response with convective component of external heat flux removed.

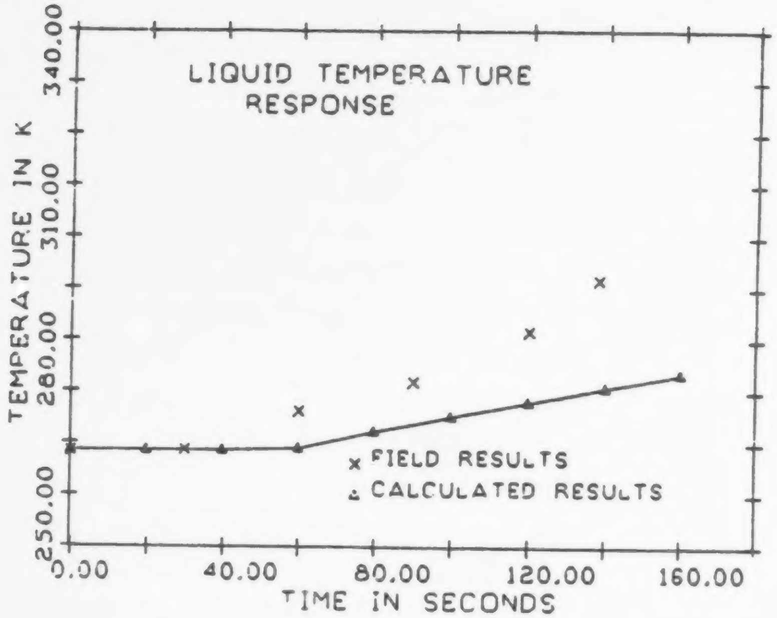


Figure 20. Bulk liquid temperature response with convective component of external heat flux removed.

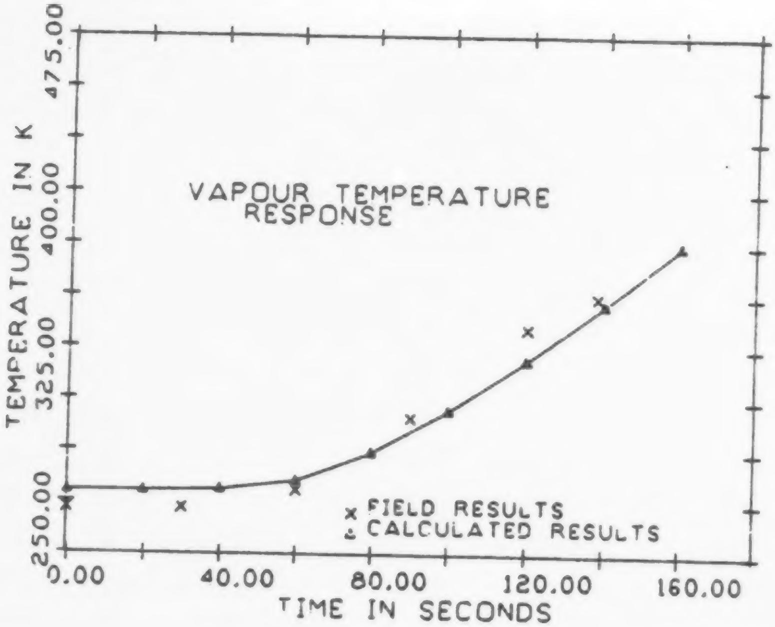


Figure 21. Bulk vapour temperature response with convective component of external heat flux removed.

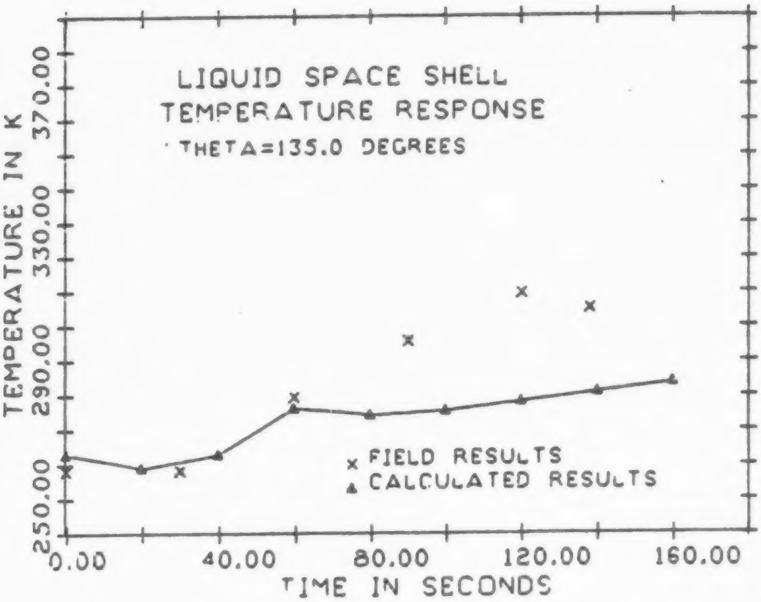


Figure 22. Liquid space shell temperature response with convective component of external heat flux removed.

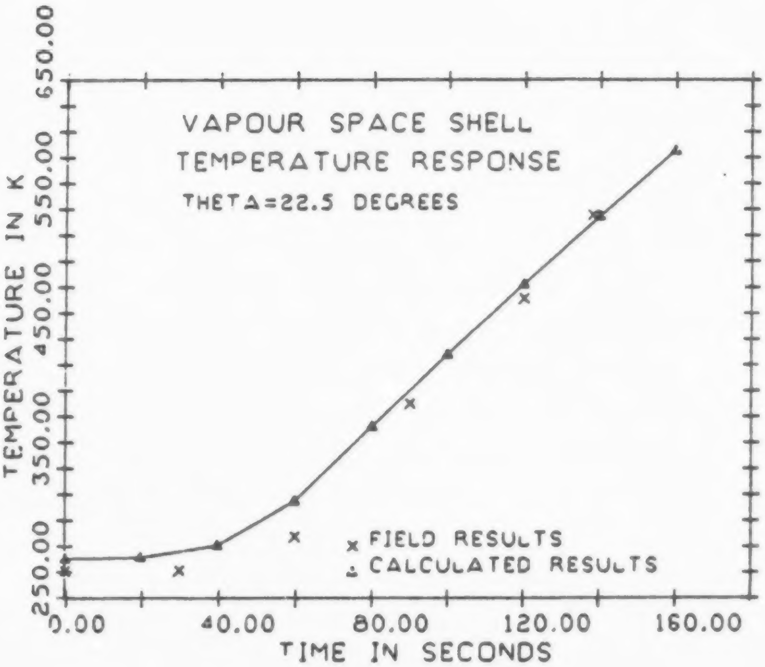


Figure 23. Vapour space shell temperature response with convective component of external heat flux removed.

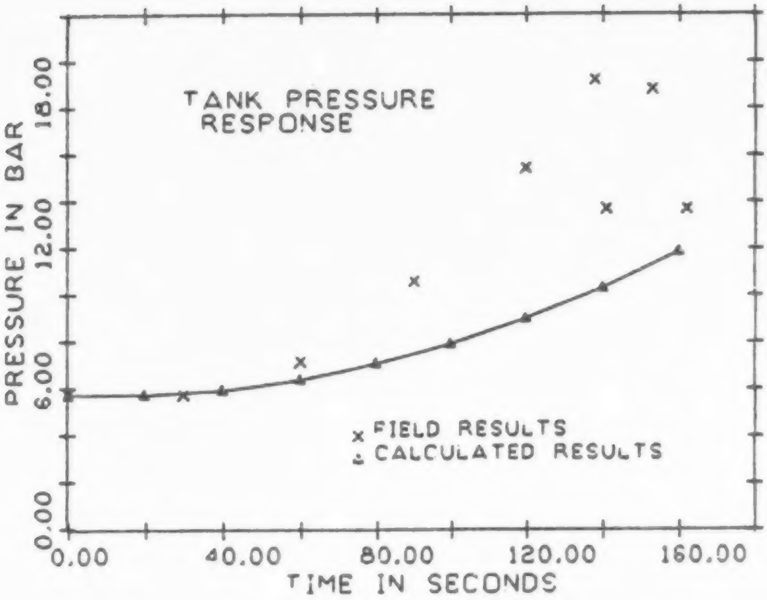


Figure 24. Internal pressure response with radiative component of external heat flux removed.

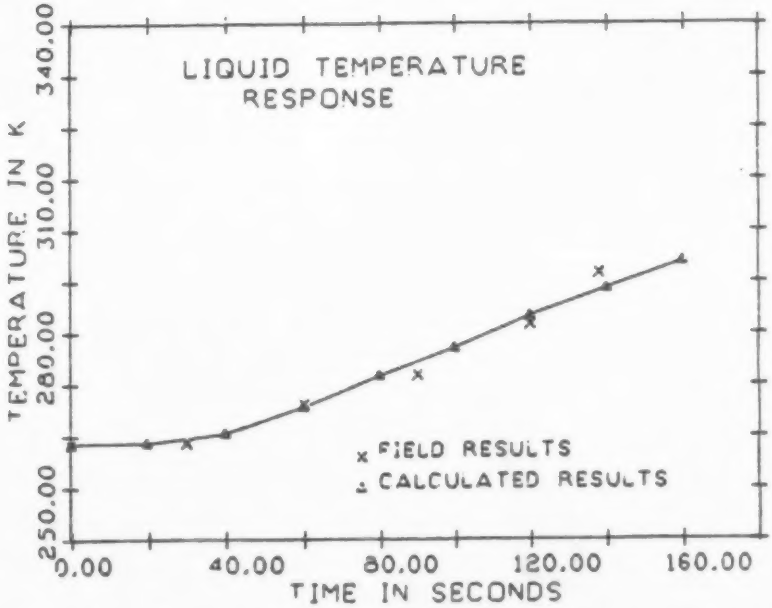


Figure 25. Bulk liquid temperature response with radiative component of external heat flux removed.

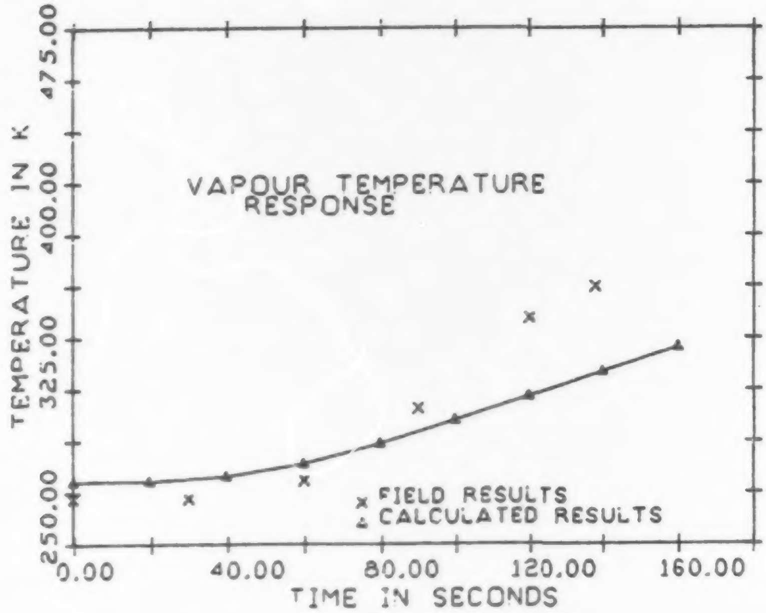


Figure 26. Bulk vapour temperature response with radiative component of external heat flux removed.

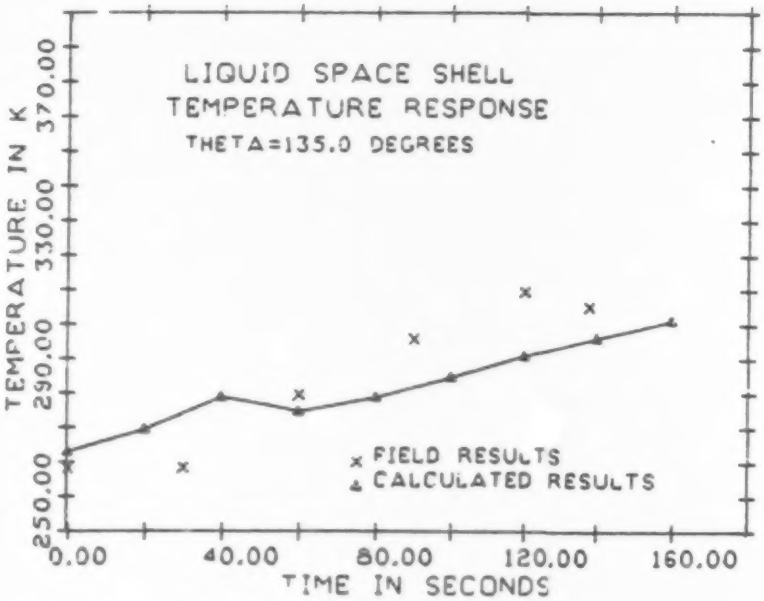


Figure 27. Liquid space shell temperature response with radiative component of external heat flux removed.

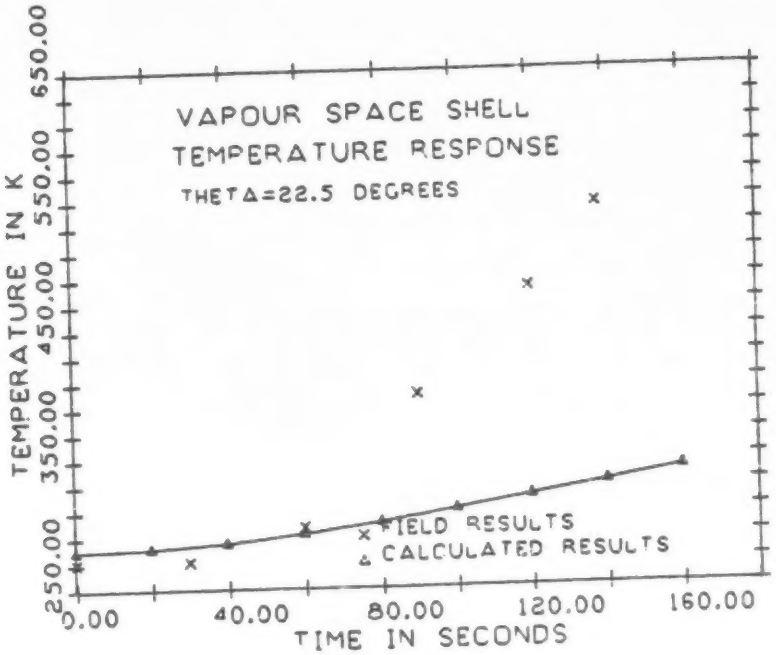


Figure 28. Vapour space shell temperature response with radiative component of external heat flux removed.

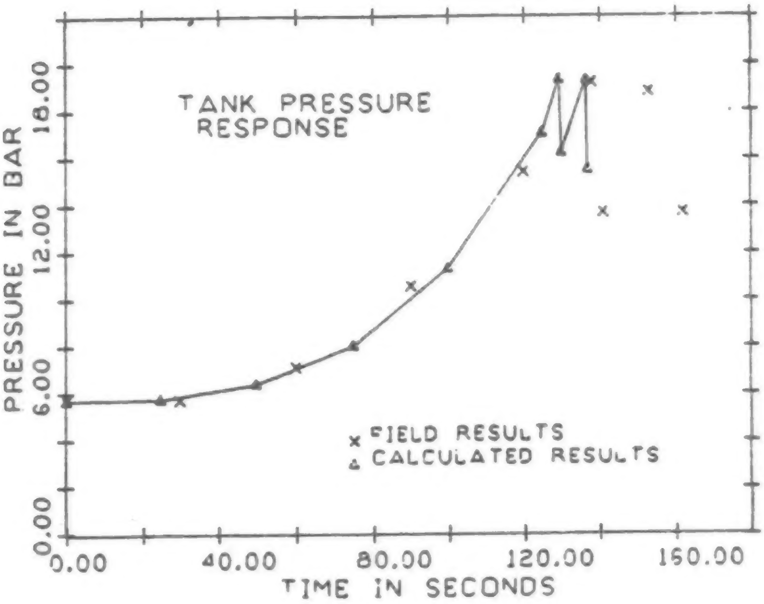


Figure 29. Internal pressure response with a total maximum heat flux of 125 kW/m^2.

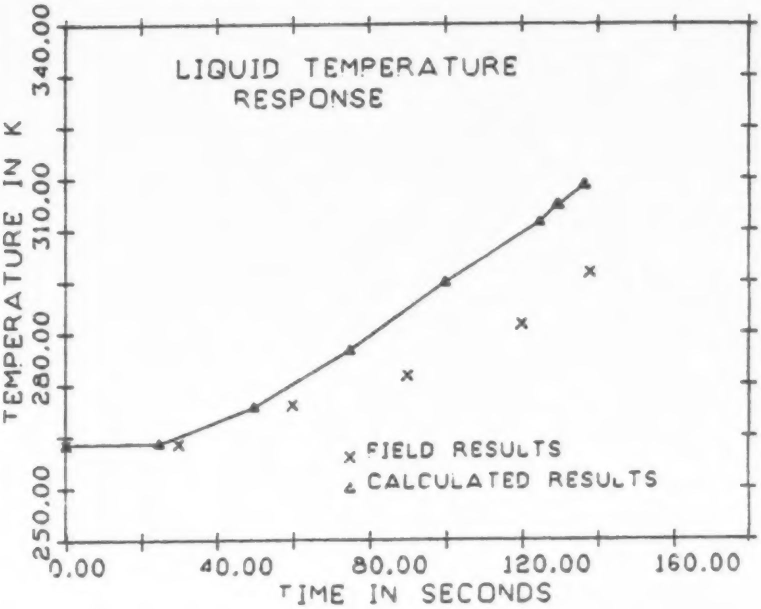


Figure 30. Bulk liquid temperature response with a total maximum heat flux of 125 kW/m^2.

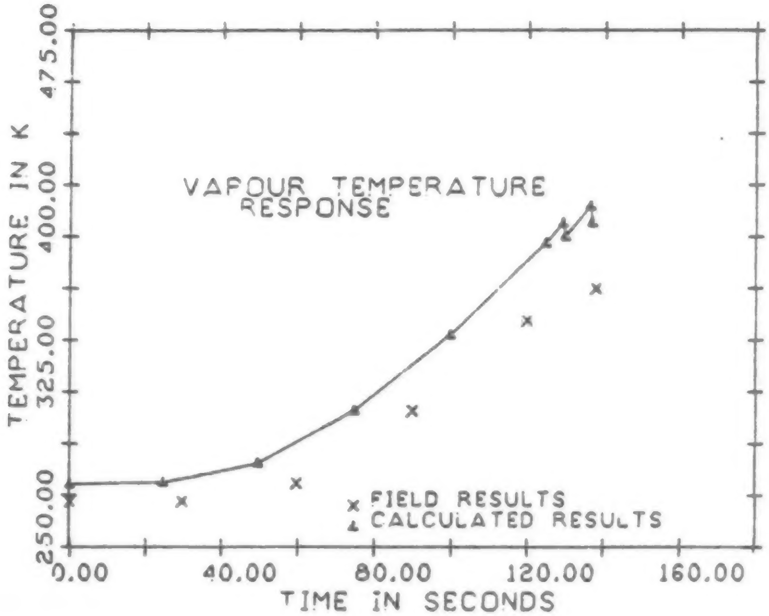


Figure 31. Bulk vapour temperature response with a total maximum heat flux of 125 kW/m^2.

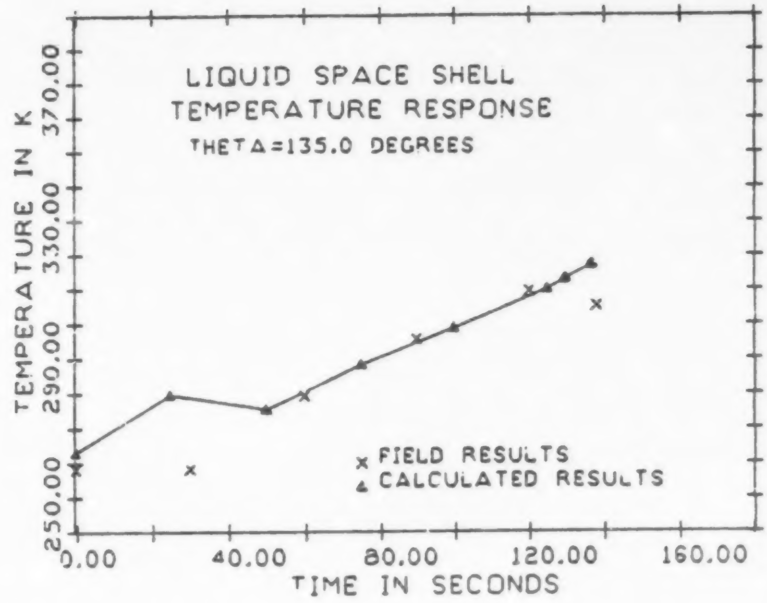


Figure 32. Liquid space shell temperature response with a total maximum heat flux of 125 kW/m^2.

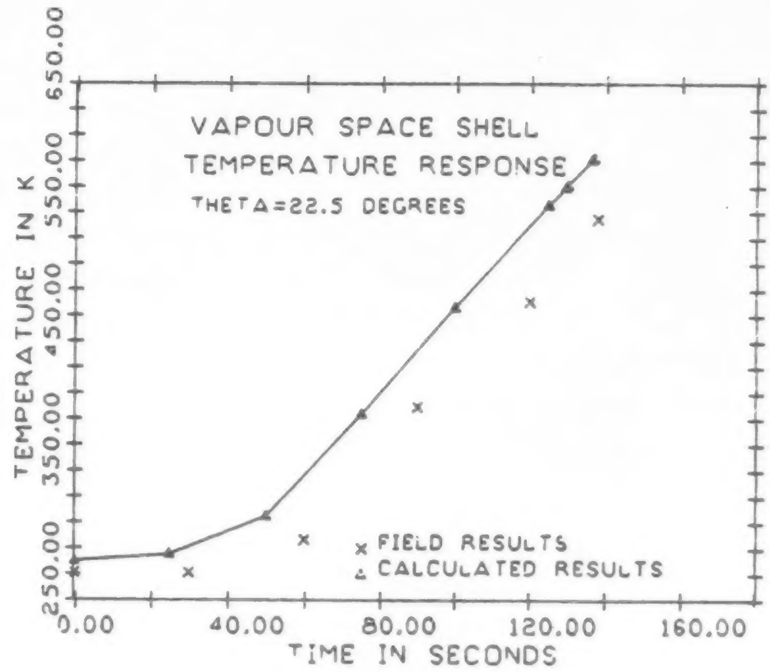


Figure 33. Vapour space shell temperature response with a total maximum heat flux of 125 kW/m^2.

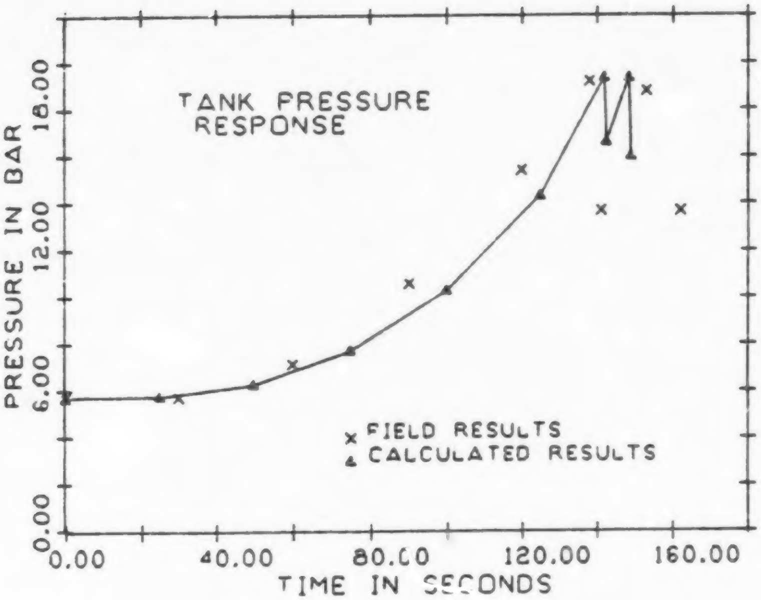


Figure 34. Internal pressure response with no vapour generation.

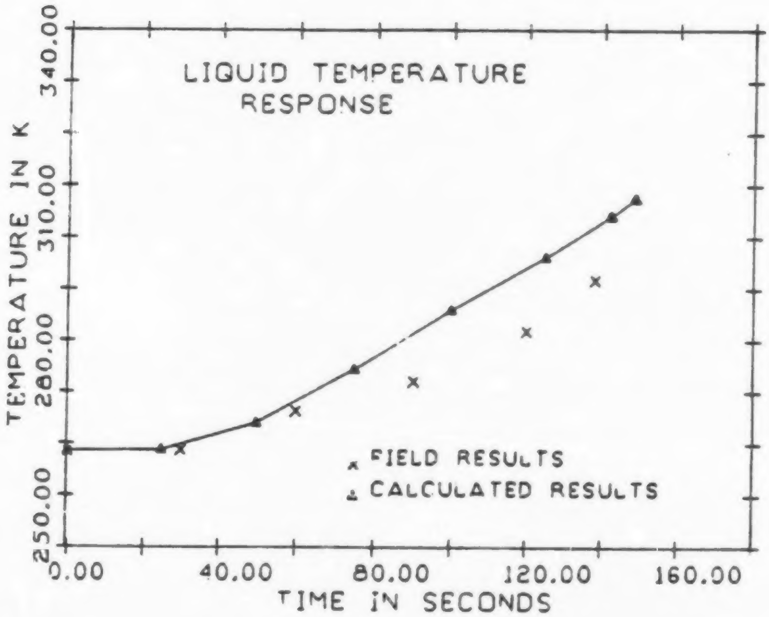


Figure 35. Bulk liquid temperature response with no vapour generation.

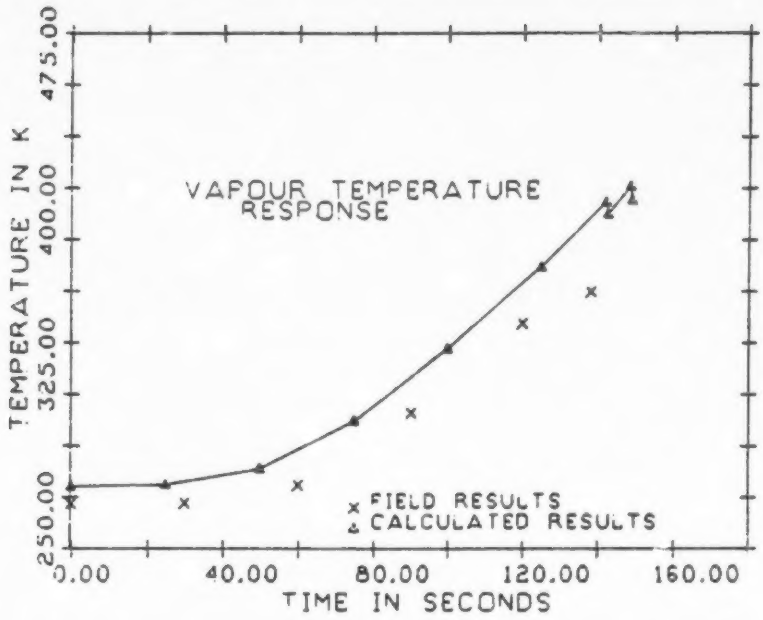


Figure 36. Bulk vapour temperature response with no vapour generation.

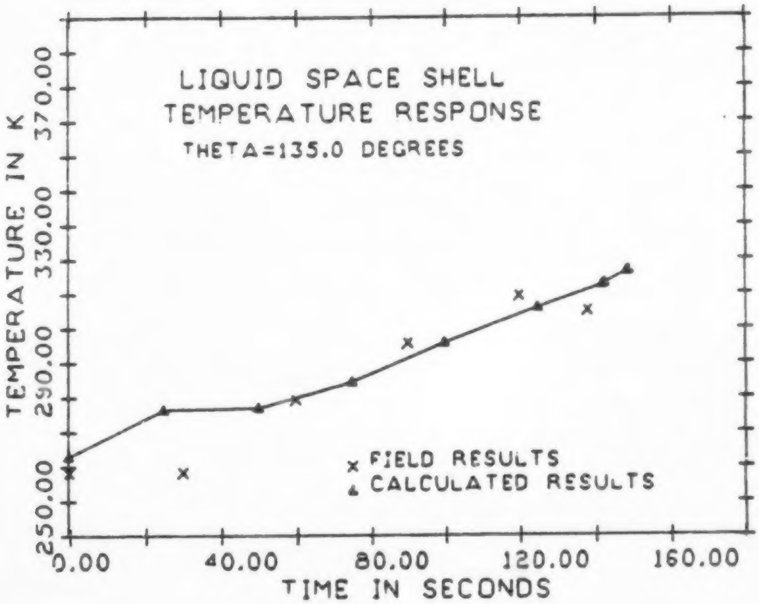


Figure 37. Liquid space shell temperature response with no vapour generation.

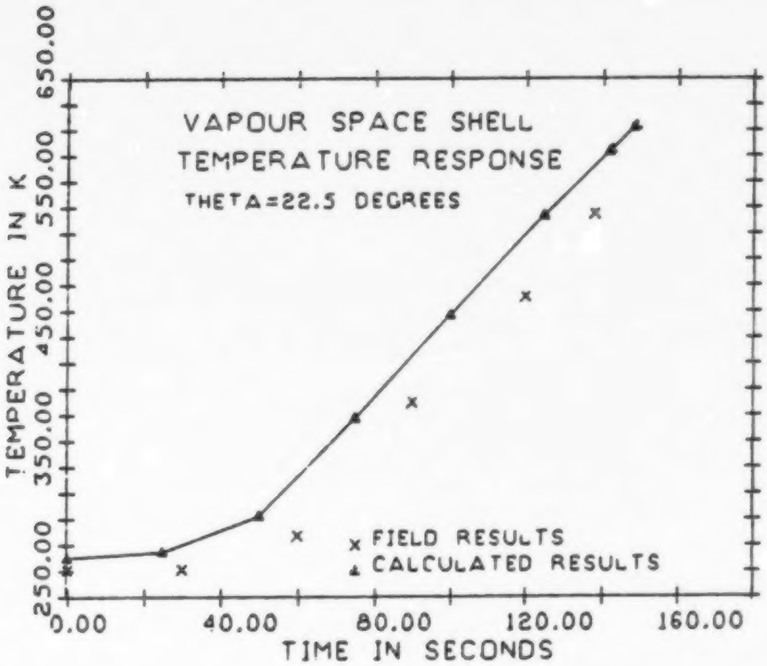


Figure 38. Vapour space shell temperature response with no vapour generation.

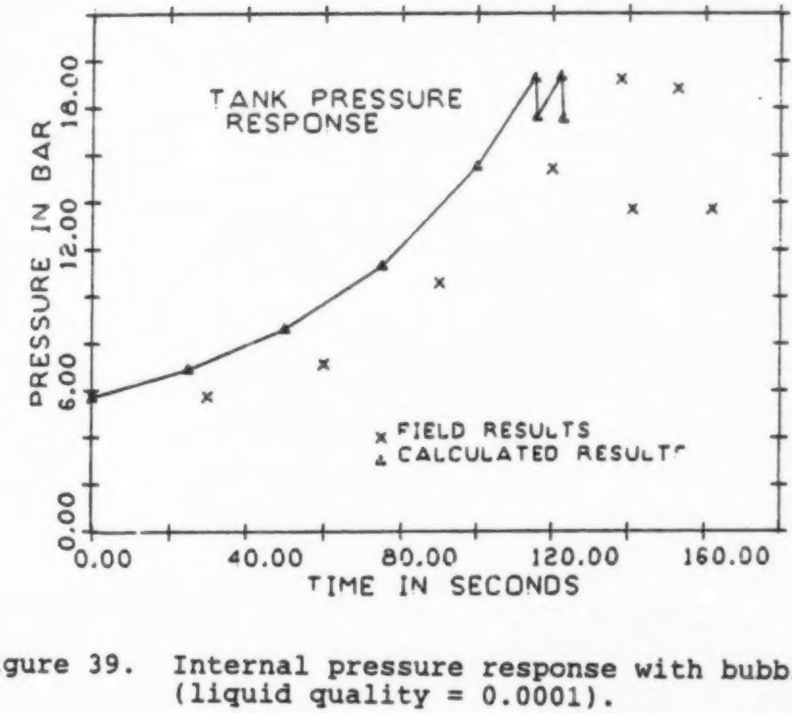


Figure 39. Internal pressure response with bubble rise (liquid quality = 0.0001).

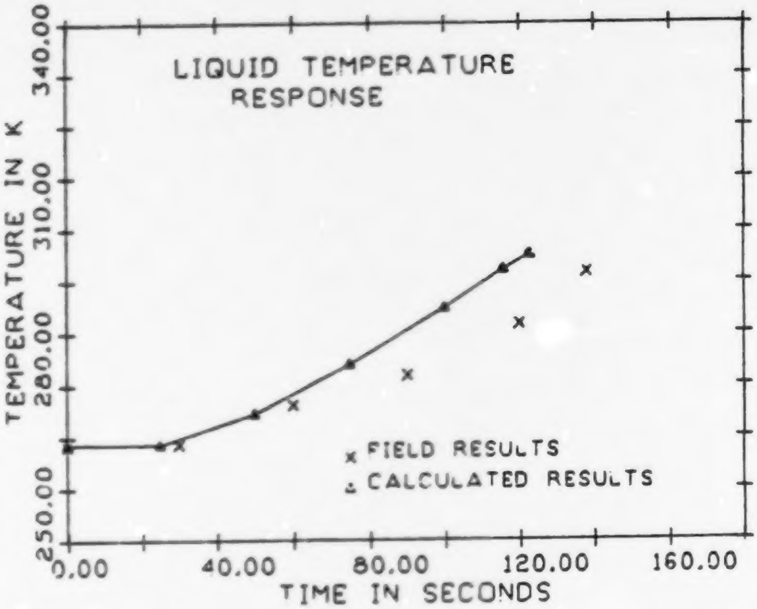


Figure 40. Bulk liquid temperature response with bubble rise (liquid quality = 0.0001).

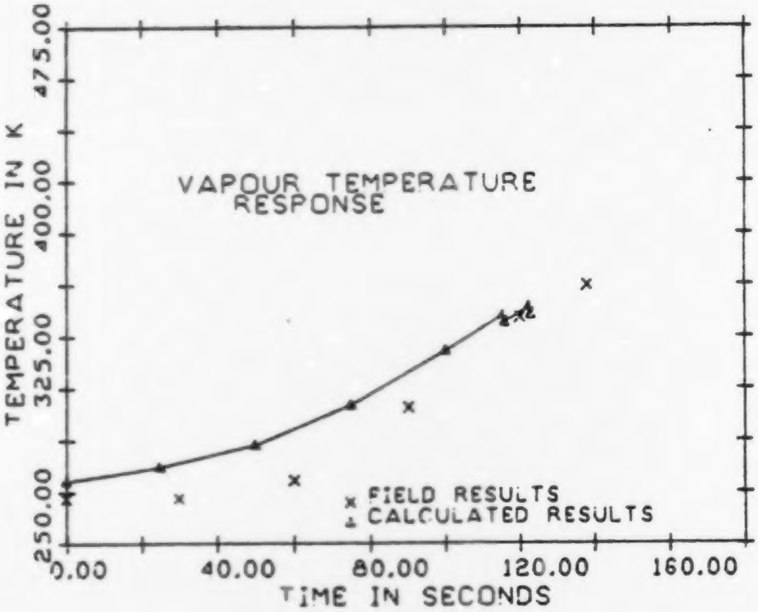


Figure 41. Bulk vapour temperature response with bubble rise (liquid quality = 0.0001).

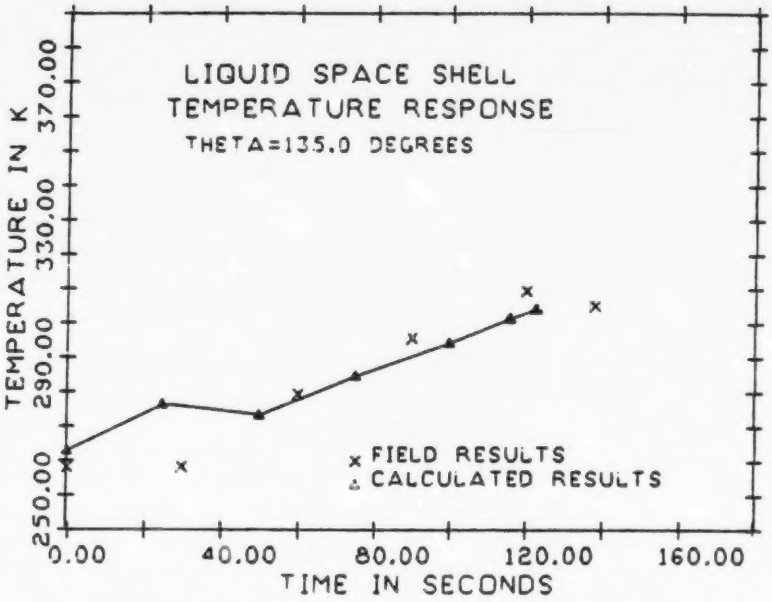


Figure 42. Liquid space shell temperature response with bubble rise (liquid quality = 0.0001).

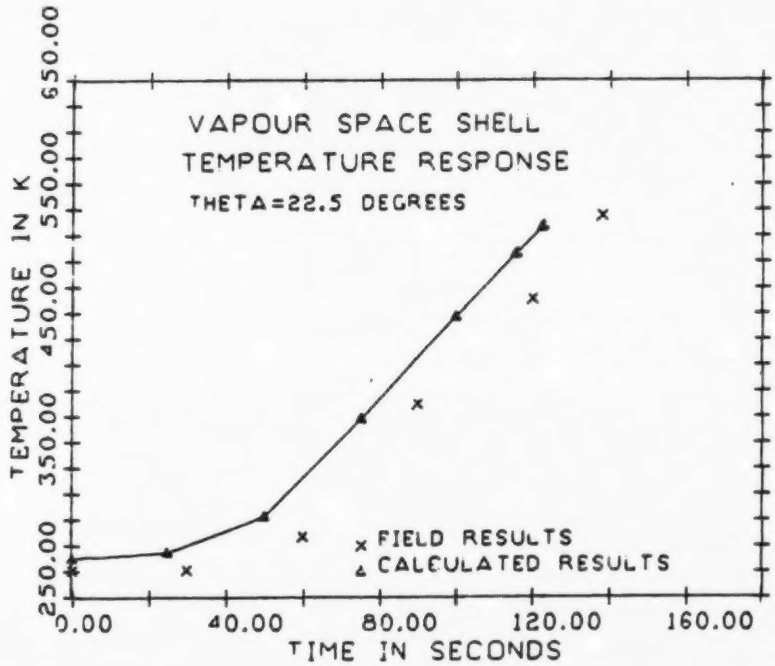


Figure 43. Vapour space shell temperature response with bubble rise (liquid quality = 0.0001).

Appendix II

COMPUTER CODING

This appendix describes the WATFIV computer coding used to determine the response of the LPG tank car subjected to a fire environment. The nomenclature used in the code is defined in the first section. The flow chart describing the basic logic of the program is then displayed followed by a brief description of the subroutines used in the model. The final section shows the computer coding with a sample output.

WATFIX Nomenclature

AREA - cross-sectional area of interface per unit depth (m)
DELTMIN - minimum time step used when changing states (s)
DELTN - normal time step size (s)
DELTT - time step used in program (s)
DELTV - time step used when pressure relief valve is open (s)
DEVAP - evaporation rate (kg/s)
DHF - rate of change of liquid enthalpy (kJ/kg s)
DHG - rate of change of vapour enthalpy (kJ/kg s)
DMBUB - bubble rise rate (kg/s)
DMF - rate of change of liquid space mass (kg/s)
DMFF - rate of change of liquid mass in liquid space (kg/s)
DMFG - rate of change of vapour mass in liquid space (kg/s)
DMG - rate of change of vapour space mass (kg/s)
DMGF - rate of change of liquid mass in vapour space (kg/s)
DMGG - rate of change of vapour mass in vapour space (kg/s)

DP - rate of change of pressure (bar/s)

DTHETA - elemental angle (radians)

EMF - initial mass in liquid space (kg)

EMFB - updated mass in liquid space (kg)

EMFF - initial mass of liquid in liquid space (kg)

EMFFB - updated mass of liquid in liquid space (kg)

EMFG - initial mass of vapour in liquid space (kg)

EMFGB - updated mass of vapour in liquid space (kg)

EMG - initial mass in vapour space (kg)

EMGB - updated mass in vapour space (kg)

EMGF - initial mass of liquid in vapour space (kg)

EMGFB - updated mass of liquid in vapour space (kg)

EMGG - initial mass of vapour in vapour space (kg)

EMGGB - updated mass of vapour in vapour space (kg)

HF - initial liquid space enthalpy (kJ/kg)

HFB - updated liquid space enthalpy (kJ/kg)

HFG - latent heat of evaporation (kJ/kg)

HFS - saturated liquid enthalpy (kJ/kg)

HG - initial vapour space enthalpy (kJ/kg)

HGB - updated vapour space enthalpy (kJ/kg)

HGS - saturated vapour enthalpy (kJ/kg)

II - time step indication

ILEVEL - nodal number at liquid/vapour interface

IOPEN - valve flag (=1: valve closed; =0: valve open)

ISTAT - initial condition of the tank lading (1-4)

KSTAT - updated condition of the tank lading (1-4)

NSTEPS - number of time steps to be executed

P - initial internal pressure (bar)

PB - updated internal pressure (bar)

PERFIL - percentage of liquid fill in the tank

PSHIGH - high pressure set of the pressure relief valve (bar)

PSLOW - low pressure set of the pressure relief valve (bar)

QF - rate of heat flow into liquid space (kJ/s)

QG - rate of heat flow into vapour space (kJ/s)

RIS - internal radius of the tank shell (m)

TAUB - mean path length of bubbles rising (m)

TF - initial liquid space temperature (K)

TFB - updated liquid space temperature (K)

TG - initial vapour space temperature (K)

TGB - updated vapur space temperature (K)

THCKS - thickness of steel shell (m)

TIME - time (s)

TN - initial shell temperature profile (K)

TNB - updated shell temperature profile (K)

TPRINT - interval time to print output (s)

TS - saturation temperature (K)

VF - liquid space specific volume (m^3/kg)

VFS - saturated liquid space specific volume (m^3/kg)

VG - vapour space specific volume (m^3/kg)

VGS - saturated vapour space specific volume (m^3/kg)

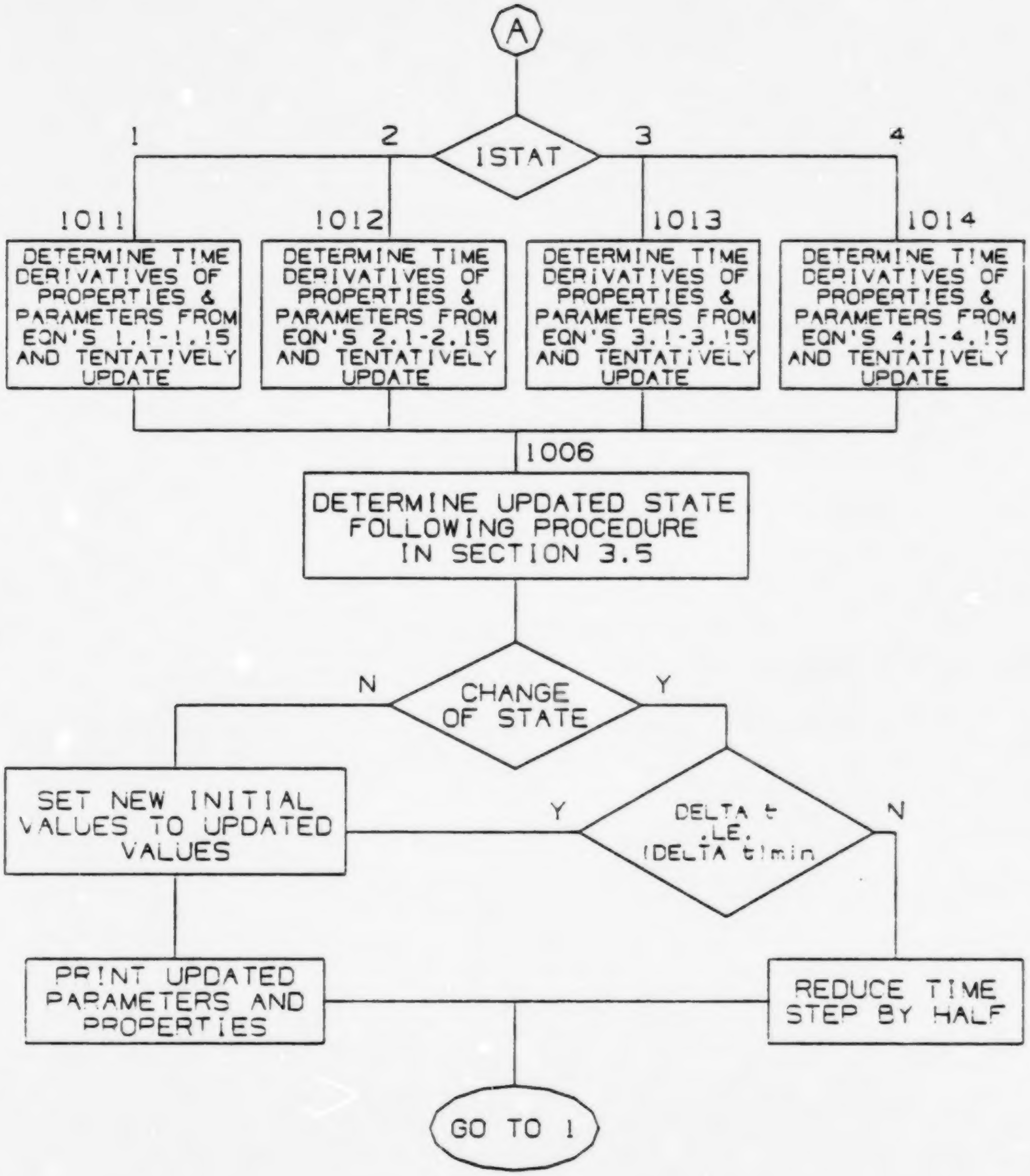
VFG - (VGS-VFS)

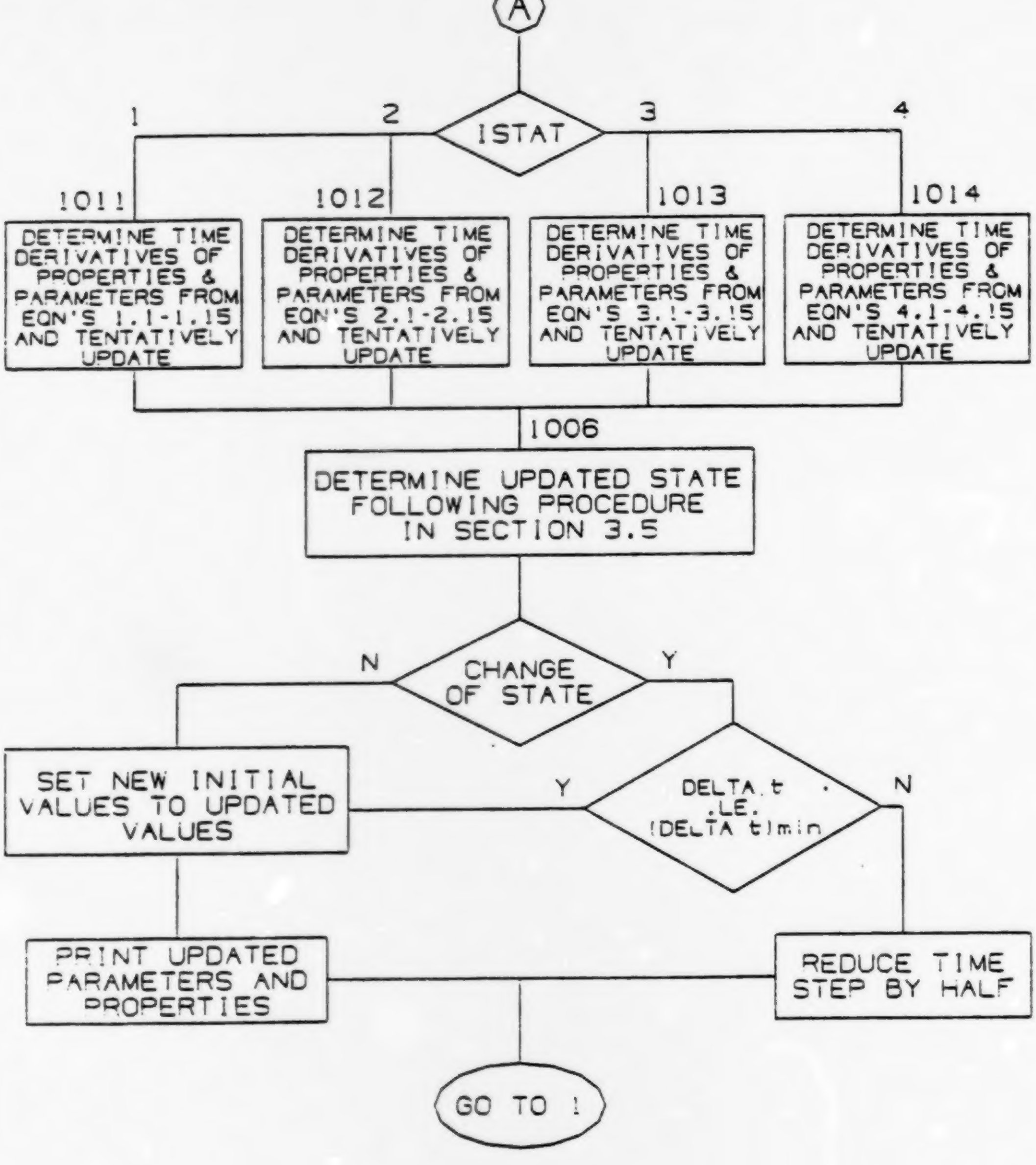
VOLF - volume of liquid space (m^3)

VOLG - volume of vapour space (m^3)

VOLT - total internal volume of tank shell (m^3)

XB - initial quality of liquid space
XBB - updated quality of liquid space
XLTH - length of tank car shell (m)
XT - initial quality of vapour space
XTB - updated quality of vapour space


A
ISTAT
1
2
3
4
1011
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 1.1-1.15 AND TENTATIVELY UPDATE
1012
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 2.1-2.15 AND TENTATIVELY UPDATE
1013
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 3.1-3.15 AND TENTATIVELY UPDATE
1014
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 4.1-4.15 AND TENTATIVELY UPDATE
1006
DETERMINE UPDATED STATE FOLLOWING PROCEDURE IN SECTION 3.5
CHANGE OF STATE
N
Y
SET NEW INITIAL VALUES TO UPDATED VALUES
Y
N
DELTA t .LE. (DELTA t)min
PRINT UPDATED PARAMETERS AND PROPERTIES
REDUCE TIME STEP BY HALF
GO TO 1


A
ISTAT
1
2
3
4
1011
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 1.1-1.15 AND TENTATIVELY UPDATE
1012
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 2.1-2.15 AND TENTATIVELY UPDATE
1013
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 3.1-3.15 AND TENTATIVELY UPDATE
1014
DETERMINE TIME DERIVATIVES OF PROPERTIES & PARAMETERS FROM EQN'S 4.1-4.15 AND TENTATIVELY UPDATE
1006
DETERMINE UPDATED STATE FOLLOWING PROCEDURE IN SECTION 3.5
CHANGE OF STATE
N
Y
SET NEW INITIAL VALUES TO UPDATED VALUES
Y
DELTA.t .LE. (DELTA t)min
N
PRINT UPDATED PARAMETERS AND PROPERTIES
REDUCE TIME STEP BY HALF
GO TO 1

Description of subroutines used in computer code:

MAIN

The bulk of the computing is done by MAIN which calls most subroutines to perform various computations. MAIN also does all of the printing and generation of results for plotting.

Calls QINTO, ILIQ, EVAP, BUBBLE, INCON, GEOM, SUBP, SUB, VDPH, SATP, BOIL, SUPP, SUP, DERIH.

Each of the program components and their associated subroutines are discussed. The program is broken down to five components: I. Initial conditions and geometrical constraints; II. Liquid level determination; III. Mass and energy fluxes; IV. Thermodyanmic properties and derivative evaluation; and V. Transport properties evaluation.

I. INITIAL CONDITIONS AND GEOMETRICAL CONSTAINTS

1. INCON

Called by: MAIN

Calls: ____

INCON inputs the initial conditions required for the program execution.

2. GEOM

Called by: MAIN

Calls: ____

GEOM set the geometrical constraints, valve data and time constraints required in the model.

II. DETERMINE THE LIQUID LEVEL

ILIQ

Called by: MAIN

Calls: ____

Determines interface cross-sectional area and bubble mean path length.

III. ENERGY AND MASS FLUXES

1. QINTO

Called by: MAIN

Calls: COEFF, HINPUT, HTSURF, TSOLCN, HTINFD

QINTO determines the amount of heat in KW which are transferred to the liquid and vapour space, through the shell.

1(a) COEFF

Called by: QINTO

Calls: ____

Determines the coefficients generated from the descretized heat conduction equation.

1(b) HINPUT

Called by: QINTO

Calls: ____

Determines the heat input to the external surface of the tank car shell due to radiation and convection from the pool fire. The theory behind HINPUT is located in the section "Flame Model".

1(c) HTSURF

Called by: QINTO

Calls: CPSL, CONDUC, VISCOS, SIGMA

Determines the liquid and vapour heat transfer coefficients on the internal surface of the tank car shell. The theory behind the development of HTSURF is located in previous sections.

1(d) TSOLCN

Called by: QINTO

Calls: ____

Determines the temperature distribution within the tank car shell, using the Crank-Nicolson technique of solution. The theory for the development of TSOLCN is located in the section "Shell Heat Transfer".

1(e) HTINFD

Called by: QINTO

Calls: ____

Determines the heat input to the lading kJ/s.

2. VENT

Called by: MAIN

Calls: ____

Determines the mass flow arte through the PRV in kg/s. The theory behind the development of VENT is located in the section "Mass Flow Rate Through the Pressure Relief Valve".

3. EVAP

Called by: MAIN

Calls: CPSL, CONDUC

Determines the evaporation rate in kg/s. The theory required for the development of EVAP is located in the section "Evaporation".

4. BUBBLE

Called by: MAIN

Calls: SIGMA

Determines the mass flow rate due to bubble rise in kg/s. The theory required for the development of BUBBLE is located in the section "Bubble rise".

IV. THERMODYNAMIC PROPERTIES AND EVALUATION OF DERIVATIVES

This component of the program determines the subcooled, saturated and superheated properties of propane and the associated derivatives for propane.

1. Subcooled Properties and Derivatives:

1(a) SUPP

Called by: MAIN

Calls: ____

Inputs subcooled property table for propane.

1(b) SUB

Called by: MAIN

Calls: XLAGR, DXLAGR

Determines subcooled properties as a function of pressure and enthalpy and also determines $\partial v/\partial h$.

1(c) VDPH

Called by: MAIN

Calls: BOIL, SUB

Determines the subcooled derivative $\partial v/\partial P$.

2. Saturated Properties and Derivatives:

2(a) SATP

Called by: MAIN

Calls: ____

Inputs saturated table values for propane.

2(b) BOIL

Called by: MAIN, VDPH

Calls: XLAGR, DXLAGR

Determines saturated properties and associated derivatives as a function of pressure.

3. Superheated Properties and Derivatives:

3(a) SUPP

Called by: MAIN

Calls: ____

Inputs superheated properties of propane.

3(b) SUP

Called by: MAIN

Calls: XLAGR, DXLAGR

Determines superheated properties and associated properties as a function of pressure and enthalpy.

3(c) DERIH

Called by: MAIN

Calls: XLAGR, DXLAGR, BOILH

Determines the derivative $\partial v/\partial h$ for superheated propane.

3(d) BOILH

Called by: DERIH

Calls: XLAGR

Determines saturated vapour enthalpy as a function of pressure but only goes to a pressure of 30 bar since there is an inflection point there.

4. Interpolation Routine

4(a) XLAGR

Called by: SUB, BOIL, SUP, DERIH, BOILH

Calls: ____

Uses a second order polynomial to fit a curve to general coordinates.

4(b) DXLAGR

Called by: SUB, BOIL, SUP, DERIH

Calls: ____

Uses a second order Lagrange polynomial to evaluate the partial derivatives of the properties.

V. TRANSPORT PROPERTIES OF PROPANE

This section deals with the determination of the transport properties of propane.

1. CPSL

   Called by: EVAP, HTSURF

   Calls: ____

   Determines the saturated liquids specific heat as a function of temperature in kJ/kmole K.

2. SIGMA

   Called by: BUBBLE, HTSURF

   Calls: ____

   Determines the surface tension of propane in dynes/cm.

3. VISCOS

   Called by: HTSURF

   Calls: ____

   Determines dynamic viscosity of propane in g/cm s.

4. CONDUC

   Called by: HTSURF, EVAP

   Calls: ____

   Determines the thermal conductivity of propane in mW/mK.

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C**********************************************************
C*     NUMERICAL SIMULATION FOR THE THERMAL RESPONSE      *
C*     OF A RAILWAY TANK CAR WHEN IT IS SUBJECTED         *
C*              TO A FIRE ENVIRONMENT                     *
C*                                                        *
C*                 RESEARCH PROJECT                       *
C*                                                        *
C*     THIS PROGRAM DETERMINES THE DYNAMIC THERMAL        *
C*      RESPONSE OF A TANK CAR LADEN WITH PROPANE         *
C*        WHEN SUBJECTED TO A FIRE ENVIRONMENT            *
C*                                                        *
C*        THE INITIAL CONDITIONS REQUIRED ARE:            *
C*         -LIQUID REGION SPECIFIC ENTHALPY               *
C*         -VAPOUR REGION SPECIFIC ENTHALPY               *
C*         -INITIAL INTERNAL PRESSURE                     *
C*         -INITIAL LIQUID PERCENTAGE FILL                *
C*                                                        *
C*            BY: P.R. WENTZELL & A.C.M. SOUSA            *
C*                         JULY 1986                      *
C**********************************************************
      IMPLICIT REAL (A-H,O-Z)
      REAL LHTCO
      REAL*8 DVDPHC,VDPH
      DIMENSION AE(21,15),AW(21,15),AN(21,15),AS(21,15),
     &APO(21,15),TN(21,15),TNB(21,15),QIN(21),QINFL(21),QOA(21),HI(21),
     &TICAL(30),TLCAL(30),TVCAL(30),PRES(30),TSHVC(30),TSHLC(30)
      COMMON / COEFFB / AE,AW,AN,AS,APO,QIN,HI
C
C     CALL THE PROPANE PROPERTY TABLES AND STORE
C
      CALL SATP
      CALL SUBP
      CALL SUPP
C
C     INITIALIZE PARAMETERS & GEOMETRIC CONSTRAINTS
C
      CALL INCON(P,HF,HG,KSTAT,PERFIL)
C
C     PRINT INITIAL CONDITIONS (I/C)
C
      PRINT 111
      PRINT 112
      PRINT 113,P,HF,HG,PERFIL
      QFT=QGT=0.0
      EFT=EGT=0.0
      IPRINT=0
      IHTCO=0
      TIME=0.0
      CALL GEOM(IOPEN,PSHIGH,PSLOW,TDIAM,RIS,XLTH,DELTN,DELTV,DTMIN,
     &DTHETA,THCKS,VOLT,NSTEPS,TPRINT)
C     INITIALIZE TIME STEPS AS NORMAL
      DELTT=DELTN
C     CALCULATE VOLUMES BASED ON I/C
      VOLF=PERFIL*VOLT
      VOLG=VOLT-VOLF
      EMF=1.0
C
```

```
C     DETERMINE STATE OF THE INITIAL CONDITIONS
C
      IF(EMF.LE.0.0)THEN DO
        PRINT,'VESSEL VOID OF LIQUID'
        GO TO 999
        END IF
      CALL BOIL(P,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
C
C     DETERMINE THE STATE
C
      IF(HG.GT.HGS)THEN DO
        IF(HF.LE.HFS)THEN DO
C
C     BOTTOM SUBCOOLED: TOP SUPERHEATED
C
          ISTAT=1
          XB=0.0
          XT=1.0
          ELSE DO
          IF(HF.LE.HGS)THEN DO
C
C     BOTTOM SATURATED: TOP SUPERHEATED
C
            ISTAT=2
C     CALCULATE FOR INITIAL CONDITIONS
            XB=(HF-HFS)/HFG
            EMF=VOLF/(VFS+XB*VFG)
            EMFG=XB*EMF
            EMFF=EMF-EMFG
            XT=1.0
            ELSE DO
            PRINT,'ERROR: BOTTOM SUPERHEATED'
            GO TO 999
            END IF
          END IF
        ELSE DO
        IF(HG.GT.HFS)THEN DO
          IF(HF.LE.HFS)THEN DO
C
C     BOTTOM SUBCOOLED: TOP SATURATED
C
            ISTAT=3
            XT=(HG-HFS)/HFG
            EMG=VOLG/(VFS+XT*VFG)
            EMGG=XT*EMG
            EMGF=EMG-EMGG
            XB=0.0
            ELSE DO
            IF(HF.LE.HGS)THEN DO
C
C     BOTTOM SATURATED: TOP SATURATED
C
              ISTAT=4
              XB=(HF-HFS)/HFG
              XT=(HG-HFS)/HFG
              EMF=VOLF/(VFS+XB*VFG)
              EMFG=XB*EMF
              EMFF=EMF-EMFG
              EMG=VOLG/(VFS+XT*VFG)
              EMGG=XT*EMG
              EMGF=EMG-EMGG
              ELSE DO
              PRINT,'ERROR: BOTTOM SUPERHEATED'
              GO TO 999
              END IF
            END IF
          ELSE DO
          PRINT,'VESSEL FULL OF LIQUID'
          END IF
        END IF
C
C     DO THE CALC FOR A NUMBER OF TIME STEPS
C
      DO 1 II=1,NSTEPS
      GO TO 13
C
C     IF CHANGE OF STATE MUST REDUCE THE TIME STEP
C
 14   DELTT=DELTT/2.0
 13   CONTINUE
      IF(ISTAT.EQ.1) GO TO 1001
      IF(ISTAT.EQ.2) GO TO 1002
      IF(ISTAT.EQ.3) GO TO 1003
      IF(ISTAT.EQ.4) GO TO 1004
 1001 CONTINUE
      CALL SUB(P,HF,TS,VFS,HFS,VF,TF,DVDHPC)
      DVDPHC=VDPH(P,HF,VF,0.05,TS,VFS,HFS)
      CALL SUP(P,HG,TS,VGS,HGS,VG,TG,DVDHPH,DVDPHH)
      GO TO 1005
 1002 CONTINUE
      CALL BOIL(P,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
C     NEED TEMPS FOR VENT AND QINTO
      TF=TS
      CALL SUP(P,HG,TS,VGS,HGS,VG,TG,DVDHPH,DVDPHH)
      GO TO 1005
 1003 CONTINUE
      CALL BOIL(P,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
C     NEED TEMPS FOR VENT AND QINTO
      TG=TS
      CALL SUB(P,HF,TS,VFS,HFS,VF,TF,DVDHPC)
      DVDPHC=VDPH(P,HF,VF,0.05,TS,VFS,HFS)
      GO TO 1005
 1004 CONTINUE
      CALL BOIL(P,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
C     NEED TEMPS FOR VENT AND QINTO
      TG=TS
      TF=TS
      GO TO 1005
 1005 CONTINUE
C
C     DETERMINE THE MASS FLOW RATE THROUGH THE P.R.V.
C
      IF(P.GE.PSHIGH)THEN DO
C     TIME STEP RESET
        DELTT=DELTV
        DMRE=VENT(P,TG,PSHIGH,PSLOW,TDIAM)
        IOPEN=0
        ELSE DO
        IF(P.GE.PSLOW)THEN DO
         IF(IOPEN.EQ.0) THEN DO
```

```
C           TIME STEP RESET
            DELTT=DELTV
            DMRE=VENT(P,TG,PSHIGH,PSLOW,TDIAM)
            IOPEN=0
            ELSE DO
            DMRE=0.0
            IOPEN=1
            END IF
          ELSE DO
          DMRE=0.0
          IOPEN=1
          END IF
        END IF
       IF(II.NE.1) GO TO 11
 11    CONTINUE
      IF(ISTAT.EQ.1) GO TO 1011
      IF(ISTAT.EQ.2) GO TO 1012
      IF(ISTAT.EQ.3) GO TO 1013
      IF(ISTAT.EQ.4) GO TO 1014
C
C     ISTAT=1: TOP SUPERHEATED; BOTTOM SUBCOOLED
C
 1011 CONTINUE
C     SET CURRENT VALUES, REQ'D IF STATE CHANGES
C     EVALUATE MASSES FOR INITIAL STATE
      IF(II.NE.1) GO TO 8
      EMF=VOLF/VF
      EMG=VOLG/VG
 8    EMFF=EMF
      EMFG=0.0
      EMGF=0.0
      EMGG=EMG
      VOLF=EMF*VF
      VOLG=EMG*VG
C
C     DETERMINE HEAT INPUT & MASS FLOW RATES
C
      CALL ILIQ(VOLG,RIS,XLTH,DTHETA,ILEVEL,WETSRF,TAUB,AREA)
      CALL QINTO(IOPEN,DELTT,VOLG,THCKS,RIS,XLTH,ILEVEL,WETSRF,II,TG,
     &TF,TS,HFG,VFS,VGS,P,TIME,ISTAT,QG,QF,TN,TNB,QOA,QINFL)
      EF=QF*DELTT
      EG=QG*DELTT
C
C     DETERMINE THE EVAPORATION RATE
C
C     INITIAL TIME STEP QF MAY BE NEGATIVE
      IF(II.EQ.1) QF=0.0
C     NO BUBBLE RISE POSS. SINCE NO VAPOUR IN LIQUID SPACE
C
        DMBUB=0.0
        DEVAP=EVAP(AREA,TN,VFS,QF,TS,TF,DELTT,HFG,ILEVEL)
C       XB=0.005
C       DMBUR=BUBBLE(VFS,VGS,EMF,XB,TAUB,TS,IOPEN)
C       DEVAP=0.0
      EMHFI=QF-DEVAP*HFG-DMBUB*HFG
      EMHGI=QG-DMRE*HG+DEVAP*HFG+DMBUB*HGS
C
C     EVALUATE THE DERIVATIVES FOR STATE 1
C
      DMF=-DEVAP-DMBUB
      DMG=-DMRE+DEVAP+DMBUB
      DP=-(DMF*VF+DMG*VG+DVDHPC*(EMHFI-DMF*HF)+DVDHPH*(EMHGI-DMG*HG))
     &/(EMF*DVDPHC+100.*VOLF*DVDHPC+EMG*DVDPHH+100.*VOLG*DVDHPH)
      DHG=(EMHGI-DMG*HG+100.*DP*VOLG)/EMG
      DHF=(EMHFI-DMF*HF+100.*DP*VOLF)/EMF
      DMFF=DMF
      DMFG=0.0
      DMGF=0.0
      DMGG=DMG
C
C     INTEGRATE THE PROPERTIES W.R.T. TIME
C
      PB=P+DP*DELTT
      HFB=HF+DHF*DELTT
      HGB=HG+DHG*DELTT
      EMFB=EMF+DMF*DELTT
      EMFFB=EMFF+DMFF*DELTT
      EMFGB=EMFG+DMFG*DELTT
      EMGB=EMG+DMG*DELTT
      EMGFB=EMGF+DMGF*DELTT
      EMGGB=EMGG+DMGG*DELTT
      GO TO 1006
C
C     ISTAT=2: TOP SUPERHEATED; BOTTOM 2-PHASE
C
 1012 CONTINUE
C     SET CURRENT VALUES, REQ'D IF STATE CHANGES
      IF(II.NE.1) GO TO 9
      EMG=VOLG/VG
 9    EMGF=0.0
      EMGG=EMG
      HF=HFS+XB*HFG
      VOLF=EMFF*VFS+EMFG*VGS
      VOLG=EMG*VG
C
C     DETERMINE HEAT INPUT & MASS FLOW RATES
C
      CALL ILIQ(VOLG,RIS,XLTH,DTHETA,ILEVEL,WETSRF,TAUB,AREA)
      CALL QINTO(IOPEN,DELTT,VOLG,THCKS,RIS,XLTH,ILEVEL,WETSRF,II,TG,
     &TF,TS,HFG,VFS,VGS,P,TIME,ISTAT,QG,QF,TN,TNB,QOA,QINFL)
      EF=QF*DELTT
      EG=QG*DELTT
C     TO ALLOW FOR CHANGING OF STATE TO 2 WHEN XB=0.0
      IF(XB.EQ.0.0) GO TO 20
C     BUBBLE RISE ONLY IF VALVE IS OPEN
      IF(ISTAT.EQ.1.OR.ISTAT.EQ.3) GO TO 20
      DMBUB=BUBBLE(VFS,VGS,EMF,XB,TAUB,TS,IOPEN)
      DEVAP=0.0
      GO TO 21
 20   DMBUB=0.0
C     EVAPORATION RATE IF VALVE IS CLOSED
      DEVAP=EVAP(AREA,TN,VFS,QF,TS,TF,DELTT,HFG,ILEVEL)
 21   EMHFI=QF-DMBUB*HGS-DEVAP*HFG
      EMHGI=QG-DMRE*HG+DMBUB*HGS+DEVAP*HGS
      EMFH=EMFF*DHFDP+EMFG*DHGDP-VOLF*100.
      EMFV=EMFF*DVFDP+EMFG*DVGDP
C
C     EVALUATE THE DERIVATIVES FOR STATE 2
C
      DMF=-DMBUB-DEVAP
      DMG=DMBUB-DMRE+DEVAP
      DP=-(HFG*(DMG*VG+DMF*VGS+DVDHPH*(EMHGI-DMG*HG))+VFG*(EMHFI-DMF
     &*HGS))/(HFG*(EMFV+EMG*DVDPHH+100.*VOLG*DVDHPH)-EMFH*VFG)
      DHG=(EMHGI-DMG*HG+100.*DP*VOLG)/EMG
      DHF=0.0
      DMFF=-(EMHFI-DMF*HGS-EMFH*DP)/HFG
      DMFG=DMF-DMFF
      DMGF=0.0
      DMGG=DMG
C
C     INTEGRATE THE PROPERTIES W.R.T. TIME
C
      PB=P+DP*DELTT
      EMFB=EMF+DMF*DELTT
      EMFFB=EMFF+DMFF*DELTT
      EMFGB=EMFG+DMFG*DELTT
      EMGB=EMG+DMG*DELTT
      EMGFB=EMGF+DMGF*DELTT
      EMGGB=EMGG+DMGG*DELTT
      HGB=HG+DHG*DELTT
C     CALCULATE HF AT THE NEW PRESSURE
      CALL BOIL(PB,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
      XBB=EMFGB/EMFB
      HFB=HFS+XBB*HFG
      GO TO 1006
C
C     ISTAT=3: TOP 2-PHASE; BOTTOM SUBCOOLED
C
 1013 CONTINUE
C     SET CURRENT VALUES, REQ'D IF STATE CHANGES
      IF(II.NE.1) GO TO 10
      EMF=VOLF/VF
 10   EMFF=EMF
      EMFG=0.0
      HG=HFS+XT*HFG
      VOLG=EMGF*VFS+EMGG*VGS
      VOLF=EMF*VF
C
C     DETERMINE HEAT INPUT & MASS FLOW RATES
C
      CALL ILIQ(VOLG,RIS,XLTH,DTHETA,ILEVEL,WETSRF,TAUB,AREA)
      CALL QINTO(IOPEN,DELTT,VOLG,THCKS,RIS,XLTH,ILEVEL,WETSRF,II,TG,
     &TF,TS,HFG,VFS,VGS,P,TIME,ISTAT,QG,QF,TN,TNB,QOA,QINFL)
      EF=QF*DELTT
      EG=QG*DELTT
C     NO BUBBLE RISE POSS. SINCE NO VAPOUR IS IN LIQUID SPACE
      DMBUB=0.0
C     EVAPORATION RATE
      DEVAP=EVAP(AREA,TN,VFS,QF,TS,TF,DELTT,HFG,ILEVEL)
      EMHFI=QF-DEVAP*HFG
      EMHGI=QG-DMRE*HG+DEVAP*HGS
      EMGH=EMGF*DHFDP+EMGG*DHGDP-100.*VOLG
      EMGV=EMGF*DVFDP+EMGG*DVGDP
C
C     EVALUATE THE DERIVATIVES FOR STATE 3
C
      DMF=-DEVAP
      DMG=-DMRE+DEVAP
      DP=-(HFG*(DMG*VGS+DMF*VF+DVDHPC*(EMHFI-DMF*HF))+VFG*(EMHGI-DMG
     &*HGS))/(HFG*(EMGV+EMF*DVDPHC+100.*VOLF*DVDHPC)-EMGH*VFG)
```

```
      DHF=(EMHFI-DMF*HF+100.*DP*VOLF)/EMF
      DHG=0.0
      DMFF=DMF
      DMFG=0.0
      DMGF=-(EMHGI-DMG*HGS-EMGH*DP)/HFG
      DMGG=DMG-DMGF
C
C     INTEGRATE THE PROPERTIES W.R.T. TIME
C
      PB=P+DP*DELTT
      EMFB=EMF+DMF*DELTT
      EMFFB=EMFF+DMFF*DELTT
      EMFGB=ENFG+DMFG*DELTT
      EMGB=EMG+DMG*DELTT
      EMGFB=EMGF+DMGF*DELTT
      EMGGB=EMGG+DMGG*DELTT
      HFB=HF+DHF*DELTT
C     CALCULATE HG AT THE NEW PRESSURE
      CALL BOIL(PB,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
      XTB=EMGGB/EMGB
      HGB=HFS+XTB*HFG
      GO TO 1006
C
C     ISTAT=4: TOP 2-PHASE; BOTTOM 2-PHASE
C
 1014 CONTINUE
C     SET CURRENT VALUES, REQ'D IF STATE CHANGES
      HG=HFS+XT*HFG
      HF=HFS+XB*HFG
      VOLG=EMGF*VFS+EMGG*VGS
      VOLF=EMFF*VFS+EMFG*VGS
C
C     DETERMINE HEAT INPUT & MASS FLOW RATES
C
      CALL ILIQ(VOLG,RIS,XLTH,DTHETA,ILEVEL,WETSRF,TAUB,AREA)
      CALL QINTO(IOPEN,DELTT,VOLG,THCKS,RIS,XLTH,ILEVEL,WETSRF,II,TG,
     &TF,TS,HFG,VFS,VGS,P,TIME,ISTAT,QG,QF,TN,TNB,QOA,QINFL)
      EF=QF*DELTT
      EG=QG*DELTT
C     IF QUALITY IS ZERO USE EVAPORATION
      IF(XB.EQ.0.0) GO TO 22
      DMBUB=BUBBLE(VFS,VGS,EMF,XB,TAUB,TS,IOPEN)
      DEVAP=0.0
      GO TO 23
 22   DMBUB=0.0
      DEVAP=EVAP(AREA,TN,VFS,QF,TS,TF,DELTT,HFG,ILEVEL)
 23   EMHFI=QF-DMBUB*HGS-DEVAP*HFG
      EMHGI=QG-DMRE*HG+DMBUB*HGS+DEVAP*HGS
      EMFH=EMFG*DHGDP+EMFF*DHFDP-100.*VOLF
      EMFV=EMFG*DVGDP+EMFF*DVFDP
      EMGH=EMGF*DHFDP+EMGG*DHGDP-100.*VOLG
      EMGV=EMGF*DVFDP+EMGG*DVGDP
C
C     EVALUATE THE DERIVATIVES FOR STATE 4
C
      DMF=-DMBUB-DEVAP
      DMG=DMBUB-DMRE+DEVAP
      DP=-(VFG*(EMHGI+EMHFI-(DMG+DMF)*HGS)+HFG*(VGS*(DMG+DMF)))
     &/(HFG*(EMGV+EMFV)-VFG*(EMGH+EMFH))
      DHF=0.0
      DHG=0.0
      DMFF=-(EMHFI-DMF*HGS-EMFH*DP)/HFG
      DMFG=DMF-DMFF
      DMGF=-(EMHGI-DMG*HGS-EMGH*DP)/HFG
      DMGG=DMG-DMGF
C
C     INTEGRATE THE PROPERTIES W.R.T. TIME
C
      PB=P+DP*DELTT
      EMFB=EMF+DMF*DELTT
      EMFFB=EMFF+DMFF*DELTT
      EMFGB=EMFG+DMFG*DELTT
      EMGB=EMG+DMG*DELTT
      EMGFB=EMGF+DMGF*DELTT
      EMGGB=EMGG+DMGG*DELTT
C     CALCULATE HF AND HG AT THE NEW PRESSURE
      CALL BOIL(PB,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
      XBB=EMFGB/EMFB
      XTB=EMGGB/EMGB
      HFB=HFS+XBB*HFG
      HGB=HFS+XTB*HFG
      GO TO 1006
 1006 CONTINUE
C
C     CHECK IF CHANGE OF STATE (IF SO REDUCE TIME STEP)
C
      IF(EMF.LE.0.0)THEN DO
        PRINT,'VESSEL VOID OF LIQUID'
        GO TO 999
        END IF
C     BOIL HAS BEEN CALLED ALREADY FOR ISTAT=2, 3, 4
      IF(ISTAT.EQ.1)THEN DO
      CALL BOIL(PB,TS,HFS,HGS,HFG,VFS,VGS,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
      END IF
C
C     DETERMINE THE STATE
C
      IF(HGB.GT.HGS)THEN DO
        IF(HFB.LE.HFS)THEN DO
C
C     BOTTOM SUBCOOLED: TOP SUPERHEATED
C
          KSTAT=1
          XBB=0.0
          XTB=1.0
          ELSE DO
          IF(HFB.LE.HGS)THEN DO
C
C     BOTTOM SATURATED: TOP SUPERHEATED
C
            KSTAT=2
C     CALCULATE FOR INITIAL CONDITIONS
            IF(ISTAT.EQ.1.OR.ISTAT.EQ.3) GO TO 15
            XBB=EMFGB/EMFB
            GO TO 16
 15         XBB=(HFB-HFS)/HFG
            IF(II.NE.1) GO TO 17
            EMFB=VOLF/(VFS+XBB*VFG)
 17         EMFGB=XBB*EMFB
            EMFFB=EMFB-EMFGB
 16         XTB=1.0
            ELSE DO
            PRINT,'ERROR: BOTTOM SUPERHEATED'
            GO TO 999
            END IF
          END IF
        ELSE DO
        IF(HGB.GT.HFS)THEN DO
          IF(HFB.LE.HFS)THEN DO
C
C     BOTTOM SUBCOOLED: TOP SATURATED
C
            KSTAT=3
            IF(II.EQ.1) GO TO 18
            XTB=EMGGB/EMGB
            GO TO 19
 18         XTB=(HGB-HFS)/HFG
            EMGB=VOLG/(VFS+XTB*VFG)
            EMGGB=XTB*EMGB
            EMGFB=EMGB-EMGGB
 19         XBB=0.0
            ELSE DO
            IF(HFB.LE.HGS)THEN DO
C
C     BOTTOM SATURATED: TOP SATURATED
C
              KSTAT=4
              IF(II.EQ.1) GO TO 24
              XBB=EMFGB/EMFB
              XTB=EMGGB/EMGB
              GO TO 25
  24          XBB=(HFB-HFS)/HFG
              XTB=(HGB-HFS)/HFG
              EMFB=VOLF/(VFS+XBB*VFG)
              EMFGB=XBB*EMFB
              EMFFB=EMFB-EMFGB
              EMGB=VOLG/(VFS+XTB*VFG)
              EMGGB=XTB*EMGB
              EMGFB=EMGB-EMGGB
  25          CONTINUE
              ELSE DO
              PRINT,'ERROR: BOTTOM SUPERHEATED'
              GO TO 999
              END IF
            END IF
          ELSE DO
          PRINT,'VESSEL FULL OF LIQUID'
          END IF
        END IF
C
      IF(KSTAT.NE.ISTAT.AND.DELTT.GT.DTMIN) GO TO 14
C
      P=PB
      HF=HFB
      HG=HGB
      EMF=EMFB
      EMFF=EMFFB
      EMFG=EMFGB
      EMG=EMGB
      EMGF=EMGFB
      EMGG=EMGGB
      XB=XBB
      XT=XTB
C
C     UPDATE FLUID TEMPERATURES
C
      IF(KSTAT.EQ.1) THEN DO
        CALL SUP(P,HG,TS,VGS,HGS,VG,TG,DVDHPH,DVDPHH)
        CALL SUB(P,HF,TS,VFS,HFS,VF,TF,DVDHPC)
        ELSE DO
        IF(KSTAT.EQ.2)THEN DO
          CALL SUP(P,HG,TS,VGS,HGS,VG,TG,DVDHPH,DVDPHH)
          ELSE DO
          IF(KSTAT.EQ.3)THEN DO
            CALL SUB(P,HF,TS,VFS,HFS,VF,TF,DVDHPC)
            END IF
          END IF
        END IF
C     UPDATE THE SHELL TEMPERATURES
      DO 31 I=2,20
        DO 31 J=5,8
          TN(I,J)=TNB(I,J)
 31       CONTINUE
      QFT=QFT+QF
      QGT=QGT+QG
      EFT=EFT+EF
      EGT=EGT+EG
      UG=HG-100.*P*VOLG/EMG
      UF=HF-100.*P*VOLF/EMF
C
C     PRINT RESULTS ONLY ONCE IN A WHILE
C
      TIME=TIME+DELTT
      TIMET=(TIME-IPRINT*TPRINT)/TPRINT
      ITIME=IFIX(TIMET)
      IF(ITIME.GT.0.OR.IOPEN.EQ.0) THEN DO
C     IF(IOPEN.EQ.0) THEN DO
      IPRINT=IPRINT+1
      PRINT,'*************************************************'
      PRINT,'*************************************************'
      PRINT,'STEP=',II
      PRINT,'DELTT=',DELTT
      PRINT 104,TIME
       DO 30 I=2,20
         PRINT 110,(TN(I,J),J=5,8),QINFL(I),QOA(I),HI(I)
 30    CONTINUE
C
C     GENERATE THE DATA FOR THE PLOTS
C
      TICAL(IPRINT)=TIME
      TLCAL(IPRINT)=TF
      TVCAL(IPRINT)=TG
      PRES(IPRINT)=P
      TSHVC(IPRINT)=TN(2,5)
      TSHLC(IPRINT)=TN(15,5)
C     PRINT THE PROPERTIES ASSOCIATED WITH EACH CONDITION
C     IF THE STATE IS 2, 3, OR 4 THIS IS UPDATED SAT. PROP.
      PRINT 101,KSTAT,XT,XB
      PRINT,'SATURATED PROPERTIES'
      PRINT,P,TS,HFS,HGS,VFS,VGS,DHFDP,DHGDP,DVFDP,DVGDP
      IF (KSTAT.EQ.1) THEN DO
        PRINT,'SUBCOOLED STATE'
        PRINT 105
        PRINT 106,P,HF,TS,VF,TF,DVDHPC,DVDPHC
        PRINT,'SUPERHEATED STATE'
        PRINT 105
        PRINT 106,P,HG,TS,VG,TG,DVDHPH,DVDPHH
        ELSE DO
        IF (KSTAT.EQ.2) THEN DO
          PRINT,'SUPERHEATED STATE'
          PRINT 105
          PRINT 106,P,HG,TS,VG,TG,DVDHPH,DVDPHH
          ELSE DO
          IF (KSTAT.EQ.3) THEN DO
            PRINT,'SUBCOOLED STATE'
            PRINT 105
            PRINT 106,P,HF,TS,VF,TF,DVDHPC,DVDPHC
            END IF
          END IF
        END IF
      PRINT,'QF=',QF,'QG=',QG
      PRINT,'QFT=',QFT,'QGT=',QGT
      PRINT,'EFT=',EFT,'EGT=',EGT
      PRINT,'UF=',UF,'UG=',UG
      PRINT,'TF=',TF,'TG=',TG
      PRINT 108,VOLF,VOLG,VOLT
      PRINT 107
      PRINT 102
      PRINT 103,P,HF,HG,EMF,EMFF,EMFG,EMG,EMGF,EMGG
      PRINT 109,DP,DHF,DHG,DMF,DMFF,DMFG,DMG,DMGG,DMGF
      END IF
      IF(ISTAT.NE.KSTAT) DELTT=DELTN
      IF(IOPEN.EQ.0) DELTT=DELTV
      ISTAT=KSTAT
 1    CONTINUE
C     PRINT OUT PLOTTING DATA
      DO 32 I=1,IPRINT
      PRINT 114,TICAL(I),PRES(I),TLCAL(I),TVCAL(I),TSHVC(I),TSHLC(I)
 32   CONTINUE
 101  FORMAT(1X,'KSTAT=',I1,3X,'XT=',F9.7,3X,'XB=',F9.7)
 102  FORMAT(2X,'BAR',5X,'KJ/KG',5X,'KJ/KG',8X,'KG',8X,'KG',11X,
     &'KG',8X,'KG',8X,'KG',8X,'KG')
 103  FORMAT (1X,F6.3,2X,F8.3,2X,F8.3,3X,F8.3,3X,F8.3,5X,F7.3,
     &3X,F7.3,3X,F7.3,3X,F7.3)
 104  FORMAT (10X,'THE TIME IS',F6.2,1X,'SECONDS')
 105  FORMAT (3X,'BAR',6X,'KJ/KG',8X,'K',6X,'CUM/KG',7X,'K',7X
     &,'CUM/KJ',6X,'CUM/(KGBAR)')
 106  FORMAT (2X,F6.3,2X,F8.3,2X,F7.3,4X,F10.7,2X,F7.3,2X,F10.7,
     &2X,F10.7)
 107  FORMAT (2X,'PRES',7X,'HF',7X,'HG',8X,'EMF',7X,'EMFF',9X,
     &'EMFG',7X,'EMG',6X,'EMGF',6X,'EMGG')
 108  FORMAT (/,2X,'VOLF=',F8.4,2X,'VOLG=',F8.4,2X,'VOLT=',F8.4)
 109  FORMAT (1X,F10.4,2X,F10.4,2X,F10.4,2X,F10.4,2X,F10.4,2X,
     &        F10.4,2X,F10.4,2X,F10.4,2X,F10.4)
 110  FORMAT (10X,7F10.2)
 111  FORMAT (/8X,'******** INITIAL CONDITIONS *******')
 112  FORMAT (6X,'PRES',7X,'HF',10X,'HG',5X,'% FILL')
 113  FORMAT (5X,F6.3,3X,F8.3,3X,F9.3,3X,F6.3)
 114  FORMAT (1X,F7.3,3X,F9.6,3X,F8.4,3X,F8.4,3X,F8.4,3X,F8.4)
      GO TO 9999
 999  PRINT,'ERROR IN MAIN'
 9999 CONTINUE
      STOP
      END
C   *********************************************
C   * THIS PORTION OF THE PROGRAM USES VARIOUS   *
C   * SUBROUTINES TO DETERMINE THE TEMPERATURE   *
C   * PROFILE OF THE SHELL AND THE HEAT TRANSFER *
C   *         INTO EACH CONTROL VOLUME          *
C   *********************************************
C     DTHETA MUST BE DEFINED IN BOTH THE MAIN PROGRAM AND
C     THE SUBROUTINE COEFFB.
C     *******************************************************
      SUBROUTINE QINTO(IOPEN,DELTT,VOLG,THCKS,RIS,XLTH,ILEVEL,WETSRF,II,
     &TGB,TFB,TSM,HFGM,VFSM,VGSM,PM,TIME,ISTAT,QV,QL,TN,TNB,QOA,QINFL)
C     *******************************************************
      IMPLICIT REAL (A-H,O-Z)
      REAL LHTCO
      DIMENSION AE(21,15),
     &AW(21,15),AN(21,15),AS(21,15),APO(21,15),
     &TNB(21,15),TN(21,15),QIN(21),HI(21),QOA(21),QINFL(21)
      COMMON / COEFFB / AE,AW,AN,AS,APO,QIN,HI
      COMMON / INDEX / NBEG,NTOT,NANG
      IND=1
      CALL COEFF(IND,II,RIS,THCKS,DELTT)
      NTOTP1=NTOT+1
      NBEGM1=NBEG-1
      NANGP1=NANG+1
      IF(II.EQ.1)THEN DO
      DO 88 J=NBEGM1,NTOTP1
        DO 88 I=1,NANGP1
          IF(I.EQ.1.OR.I.EQ.NANGP1.OR.J.EQ.NBEGM1.OR.J.EQ.NTOTP1)THEN DO
            TNB(I,J)=0.0
            TN(I,J)=0.0
            ELSE DO
C
C   CAN ONLY DO THIS THE FIRST TIME STEP
C
            IF(I.LT.ILEVEL) TINF=TGB
C           TO TAKE INTO ACCOUNT THE INTERFACE ELEMENT BY AVERAGE
            IF(I.EQ.ILEVEL) TINF=(TGB+TFB)/2.
            IF(I.GT.ILEVEL) TINF=TFB
            TNB(I,J)=TINF
            TN(I,J)=TINF
            END IF
 88       CONTINUE
        END IF
C     BASED ON TEMP. AT BEGINNING OF TIME STEP
      CALL HINPUT(TN,TIME,QINFL)
C
C     PRINT INITIAL HEAT FLUX
C
      IF(II.NE.1)GO TO 9
      PRINT 104
      DO 8 I=2,NANG
        PRINT 103,QIN(I),I
        QOA(I)=1.
  8     CONTINUE
 201  CONTINUE
C     BASED ON TEMP. AT BEGINNING OF TIME STEP
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  9   CALL HTSURF(IOPEN,ILEVEL,WETSRF,QOA,TIME,ISTAT,II,TN,PM,TSM,TFB,
     &HFGM,VGSM,VFSM,RIS,VHTCO,LHTCO,HI)
 14   CALL TSOLCN(II,TGB,TFB,ILEVEL,TN,TNB)
C       BASED ON TEMP. AT BEGINNING OF TIME STEP
        CALL HTINFD(VOLG,XLTH,TGB,TFB,ILEVEL,TN,QV,QL,QOA)
 103  FORMAT (10X,F10.3,5X,I3)
 104  FORMAT (12X,'QIN',8X,'INDEX')
      RETURN
      END
C
C----------------------------------------
      SUBROUTINE COEFF(IND,II,RIS,THCKS,DELTT)
C----------------------------------------
C
C     THIS SUBROUTINE DETERMINES THE COEFFICIENTS
C     REQUIRED TO DETERMEINE THE TEMP. PROFILE IN THE SHELL
C
C     DENSS- DENSITY OF THE STEEL SHELL (KG/CU.M)
C     CPS  - SPECIFIC HEAT OF THE STEEL SHELL (KJ/KG K)
C     CONDR- RADIAL CONDUCTIVITY (KW/M K)
C     CONDC- CIRCUMFERENTIAL CONDUCTIVITY (KW/M K)
C
      IMPLICIT REAL (A-H,O-Z)
      COMMON / COEFFB / AE,AW,AN,AS,APO,QIN,HI
      COMMON / MESH / THETA,DELTH,DTHETA,R,DR,DELTR
      COMMON / INDEX / NBEG,NTOT,NANG
      DIMENSION DR(15),DELTR(15),R(15),THETAD(21),THETA(21)
     &,DELTH(21),DENS(15),CP(15),CONDR(15),CONDC(15),AE(21,15),
     &AW(21,15),AN(21,15),AS(21,15),APO(21,15),
     &QIN(21),HI(21)
      DATA THCKII,THCKIO / .24 ,.24 /
      DATA NELS,NELII,NELIO / 3 ,3 ,3 /
      DATA CPII,CPS,CPIO / 480.,480.,480./
      DATA DENSII,DENSS,DENSIO / 7800.,7800.,7800. /
      DATA CONDII,CONDS,CONDIO / 50. ,50.,50. /
      DATA NANG / 20 /
C   --------------------------------------------------
C    IND=1: SHELL ONLY
C    IND=2: SHELL WITH EXTERIOR INSULATION
C    IND=3: SHELL WITH INTERIOR AND EXTERIOR INSULATION
C    IND=4: SHELL WITH INTERIOR INSULATION
C   --------------------------------------------------
      IF(II.EQ.1) PRINT,'**************','IND=',IND,'*****************'
      PI=3.141592654
      RIII=RIS-THCKII
      IF(IND.EQ.1.OR.IND.EQ.2) THEN DO
        NBEG=NELII+2
        R(NBEG-1)=RIS
        ELSE DO
        NBEG=2
        R(NBEG-1)=RIS-THCKII
        END IF
      IF(IND.EQ.2.OR.IND.EQ.3) NTOT=NELS+NELII+NELIO+2
      IF(IND.EQ.1.OR.IND.EQ.4) NTOT=NELS+NELII+2
      DR(NBEG-1)=0.0
      DR(NTOT)=0.0
      DO 1 J=NBEG,NTOT
        IF(J.EQ.NTOT) GO TO 2
        IF(J.LE.NELII+1) DR(J)=THCKII/NELII
        IF(J.GT.NELII+1.AND.J.LE.NELII+NELS+1) DR(J)=THCKS/NELS
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```
         IF(J.GT.NELII+NELS+1) DR(J)=THCKIO/NELIO
C
C        R STARTS AT 2
C
 2       CONTINUE
         R(J)=R(J-1)+DR(J-1)
         DELTR(J)=(DR(J)+DR(J-1))/2.0
         IF(II.EQ.1) PRINT,J,R(J),DR(J),DELTR(J)
 1       CONTINUE
C
C     DEFINE THE ANGLES
C
       IF(II.EQ.1) PRINT,'************ ANGLES ************'
      DO 3 I=2,NANG
         DTHETA=PI/(FLOAT(NANG)-2.0)
         THETA(I)=DTHETA*FLOAT(I-2)
         IF(I.EQ.2.OR.I.EQ.NANG) THEN DO
           DELTH(I)=DTHETA/2.0
       ELSE DO
       DELTH(I)=DTHETA
       END IF
       THETAD(I)=THETA(I)*180./PI
       IF(II.EQ.1) PRINT,I,THETAD(I),DTHETA,DELTH(I)
 3     CONTINUE
C
C     DEFINE NODAL PROPERTIES
C
      DO 4 J=NBEG,NTOT
        IF(J.LT.NELII+2) CONDR(J)=CONDII
        IF(J.GE.NELII+2.AND.J.LT.NELII+NELS+2) CONDR(J)=CONDS
        IF(J.GE.NELII+NELS+2) CONDR(J)=CONDIO
        IF(J.LT.NELII+2) THEN DO
          CONDC(J)=CONDII
          CP(J)=CPII
          DENS(J)=DENSII
          ELSE DO
          IF(J.GT.NELII+2.AND.J.LT.NELII+NELS+2) THEN DO
            CONDC(J)=CONDS
            CP(J)=CPS
            DENS(J)=DENSS
            ELSE DO
            IF(J.GT.NELII+NELS+2) THEN DO
              CONDC(J)=CONDIO
              CP(J)=CPIO
              DENS(J)=DENSIO
              END IF
            END IF
          END IF
 4      CONTINUE
     IF(II.EQ.1) PRINT,'******* NODAL PROPERTIES *******'
      DO 5 J=NBEG,NTOT
       IF(J.EQ.NELII+2.OR.J.EQ.NELII+NELS+2) THEN DO
       IF(IND.EQ.1) THEN DO
          CONDC(NELII+2)=CONDS
          DENS(NELII+2)=DENSS
          CP(NELII+2)=CPS
          CONDC(NELII+NELS+2)=CONDS
          DENS(NELII+NELS+2)=DENSS
          CP(NELII+NELS+2)=CPS
          GO TO 6
          ELSE DO
          IF(IND.EQ.2) THEN DO
          CONDC(NELII+2)=CONDS
          DENS(NELII+2)=DENSS
          CP(NELII+2)=CPS
          IF(J.EQ.NELII+2) GO TO 6
          ELSE DO
          IF(IND.EQ.4) THEN DO
            CONDC(NELII+NELS+2)=CONDS
            DENS(NELII+NELS+2)=DENSS
            CP(NELII+NELS+2)=CPS
            IF(J.EQ.NELII+NELS+2) GO TO 6
            END IF
          END IF
        END IF
        DUM1=(R(J)-DR(J-1)/4.0)*DENS(J-1)*DR(J-1)
        DUM2=(R(J)+DR(J)/4.0)*DENS(J+1)*DR(J)
        DENS(J)=(DUM1+DUM2)/(2.0*R(J)*DELTR(J))
        CP(J)=(DUM1*CP(J-1)+DUM2*CP(J+1))/(DUM1+DUM2)
        CONDC(J)=(DR(J-1)*CONDC(J-1)+DR(J)*CONDC(J+1))/(2.0*DELTR(J))
        END IF
 6      IF(II.EQ.1) PRINT,J,DENS(J),CP(J),CONDR(J),CONDC(J)
 5      CONTINUE
C
C     GENERATE THE COEFFICIENTS
C
      DO 7 J=NBEG,NTOT
        DO 7 I=2,NANG
          APO(I,J)=DENS(J)*CP(J)*R(J)*DELTR(J)*DELTH(I)/DELTT
C     AE
          IF(I.EQ.NANG) AE(I,J)=0.0
          IF(I.NE.NANG) AE(I,J)=CONDC(J)*DELTR(J)/(R(J)*DTHETA)
C         MUST GO ONLY TO NTOT-1
C     AW
          IF(I.EQ.2) AW(I,J)=0.0
          IF(I.NE.2) AW(I,J)=CONDC(J)*DELTR(J)/(R(J)*DTHETA)
C     AN
          IF(J.EQ.NTOT) AN(I,J)=0.0
          IF(J.LT.NTOT.AND.J.GT.NBEG) AN(I,J)=CONDR(J)*(R(J)+DELTR(J)
     &                                      /2.0)*DELTH(I)/DR(J)
          IF(J.EQ.NBEG) AN(I,J)=CONDR(J)*(R(J+1)-DELTR(J+1)/2.0)
     &                     *DELTH(I)/DR(J)
C     AS
          IF(J.EQ.NBEG) AS(I,J)=0.0
          IF(J.EQ.NTOT) AS(I,J)=CONDR(J-1)*(R(J-1)+DELTR(J-1)/2.0)*
     &                      DELTH(I)/DR(J-1)
          IF(J.GT.NBEG.AND.J.LT.NTOT) AS(I,J)=CONDR(J-1)*(R(J-1)+
     &                            DELTR(J-1)/2.0)*DELTH(I)/DR(J-1)
 7        CONTINUE
      RETURN
      END
C
C
C-----------------------------------
      SUBROUTINE TSOLCN(II,TGB,TFB,ILEVEL,TN,TNB)
C-----------------------------------
C
C     THIS SUBROUTINE DETERMINES THE TEMP PROFILE
C     OF THE TANK SHELL IMPLICITLY
C
```

```
C       II=   TIME COUNTER
C       TNB=   NEW TEMPERATURE PROFILE IN K
C       TN =   PREVIOUS TEMP. PROFILE IN K
C       HI=   INTERNAL HEAT TRANS. COEFF. IN W/SQMK
C       TGB= BULK TEMP. OF VAPOUR
C       TFB=BULK TEMP. OF LIQUID
C
      IMPLICIT REAL (A-H,O-Z)
      COMMON / COEFFB / AE,AW,AN,AS,APO,QIN,HI
      COMMON / MESH / THETA,DELTH,DTHETA,R,DR,DELTR
      COMMON / INDEX / NBEG,NTOT,NANG
      DIMENSION DR(15),DELTR(15),R(15),THETA(21)
     &,DELTH(21),AE(21,15),
     &AW(21,15),AN(21,15),AS(21,15),APO(21,15),B(21,15),
     &TNB(21,15),TN(21,15),QIN(21),HI(21),AP(21,15),
     &HTIN(21,15),CONV(21,15)
      F=0.5
      ITMAX=80
      EPSLON=0.001
      ITER=0
      NTOTP1=NTOT+1
      NBEGM1=NBEG-1
      NANGP1=NANG+1
      DO 13 J=NBEG,NTOT
        DO 13 I=2,NANG
          IF(J.EQ.NBEG) THEN DO
            CONV(I,J)=HI(I)*R(J)*DELTH(I)
            HTIN(I,J)=0.0
            ELSE DO
            IF(J.EQ.NTOT) THEN DO
              CONV(I,J)=0.0
              HTIN(I,J)=QIN(I)
             ELSE DO
              CONV(I,J)=0.0
              HTIN(I,J)=0.0
              END IF
            END IF
C         EXTRA REQUIRED VARIABLES ON ANOTHER PROGRAM
        AP(I,J)=APO(I,J)+F*(AE(I,J)+AW(I,J)+AN(I,J)+AS(I,J)+CONV(I,J))
 13       CONTINUE
      DO 17 J=NBEG,NTOT
        DO 17 I=2,NANG
          IF(I.LT.ILEVEL) TINF=TGB
C         TO AVERAGE THE INTERFACE ELEMENT
          IF(I.EQ.ILEVEL) TINF=(TFB+TGB)/2.
          IF(I.GT.ILEVEL) TINF=TFB
C         SOURCE TERM BASED ON TEMP. AT BEGINNING OF TIME STEP
          B(I,J)=HTIN(I,J)+CONV(I,J)*TINF+(APO(I,J)-(1.0-F)*(AE(I,J)+
     &              AW(I,J)+AN(I,J)+AS(I,J)+CONV(I,J)))*TN(I,J)+(1.0-F)*
     &              (AE(I,J)*TN(I+1,J)+AW(I,J)*TN(I-1,J)+AN(I,J)*TN(I,J+1)
     &              +AS(I,J)*TN(I,J-1))
17      CONTINUE
200     CONTINUE
      RES=0.0
      ITER=ITER+1
      DO 11 J=NBEG,NTOT
        DO 11 I=2,NANG
          TO=TNB(I,J)
          TNB(I,J)=(F*(AE(I,J)*TNB(I+1,J)+AW(I,J)*TNB(I-1,J)
     &             +AN(I,J)*TNB(I,J+1)+ AS(I,J)*TNB(I,J-1))+B(I,J))/AP(I,J)
```

```
            RES1=ABS(TNB(I,J)-TO)
            IF(RES1.GT.RES) RES=RES1
 11       CONTINUE
      IF(ITER.GT.ITMAX)GO TO 15
      IF(RES.LE.EPSLON) GO TO 16
      GO TO 200
 15   PRINT,'NO CONVERGENCE'
      PRINT,'RES=',RES
 16   CONTINUE
      RETURN
      END
C
C-------------------------------------
      SUBROUTINE HINPUT(TN,TIME,QINFL)
C-------------------------------------
C
C     THIS SUBROUTINE DETERMINES THE HEAT INPUT
C     TO THE EXTERNAL SURFACE OF THE TANK
C
C     QIN     = HEAT INPUT IN W / UNIT LENGTH
C     POWER   = QIN/A IN KW/SQM
C     ANGLE   = ANGLE FROM T.D.S. IN RAD
C     R(NTOT)= OUTSIDE RADIUS IN M
C     TFMAX =MAX. FIRE TEMP. IN K
C     TBASE =INITIAL FIRE TEMP. IN K
C     TFIRE = TIME FOR FIRE TEMP. TO REACH MAXIMUM
      IMPLICIT REAL (A-H,O-Z)
      COMMON / COEFFB / AE,AW,AN,AS,APO,QIN,HI
      COMMON / MESH / THETA,DELTH,DTHETA,R,DR,DELTR
      COMMON / INDEX / NBEG,NTOT,NANG
      DIMENSION DR(15),DELTR(15),R(15),THETA(21),DELTH(21),AE(21,15),
     &AW(21,15),AN(21,15),AS(21,15),APO(21,15),QIN(21),HI(21),TN(21,15),
     &QINFL(21)
C
C     NHALF= INDICE AT 90
      NHALF=NANG/2
      TFMAX=870.+273.
      TBASE=15.5+273.
      TB2=TBASE*TBASE
      TB4=TB2*TB2
      TFM2=TFMAX*TFMAX
      TFM4=TFM2*TFM2
C     FIRE TEMP. HISTORY
      TIFIRE=60.
      IF(TIME.LT.TIFIRE) THEN DO
C     TIME + 1. SEC IS USED TO ACCOUNT FOR INITIAL TIME STEP
        TFIRE=TBASE+(TFMAX-TBASE)*(TIME+1.)/TIFIRE
        ELSE DO
        TFIRE=TFMAX
        END IF
      TF2=TFIRE*TFIRE
      TF4=TF2*TF2
      DO 1 I=2,NANG
C     WALL TEMPS.
        TW2=TN(I,NTOT)*TN(I,NTOT)
        TW4=TW2*TW2
        ANGLE=THETA(I)+DELTH(I)/2.0
C
C    MAXIMUM HEAT FLUX INPUT @ DELTA TEMP=TFIREMX-15.5
C
```

```
      IF(I.GT.NHALF) THEN DO
        PMAXT=108.8
        ELSE DO
        PMAXT=108.8-(108.8-70.1784)*FLOAT(NHALF-I)/FLOAT(NHALF)
        END IF
C
C     RADIATION FROM BIRK'S MODEL
C
        PMAXR=70.1784+(-12.4548+(-4.97278+(1.35473*ANGLE))*
     &        ANGLE)*ANGLE
C
C       CONVECTIVE COMPONENT
C
        PMAXC=PMAXT-PMAXR
        POWER=PMAXR*(TF4-TW4)/(TFM4-TB4)+PMAXC*(TFIRE-TN(I,NTOT))/
     &        (TFMAX-TBASE)
        QIN(I)=POWER*R(NTOT)*DELTH(I)*1.0E03
        QINFL(I)=POWER*1.0E3
 1      CONTINUE
      RETURN
      END
C
C---------------------------------------------------------------------
      SUBROUTINE HTSURF(IOPEN,ILEVEL,WETSRF,QOA,TIME,ISTAT,II,TW,PM,TSM,
     &TFM,HFGM,VGM,VFM,RIS,VHTCO,LHTCO,HI)
C---------------------------------------------------------------------
C
C    THIS SUBROUTINE DETERMINES THE HEAT TRANSFER
C     COEFFICIENTS ON THE INTERNAL SHELL SURFACE
C
C     PM=    INTERNAL PRESSURE IN BAR
C     PE=    INTERNAL PRESSURE IN PSIA
C     TW=    INSIDE WALL TEMP. @ BEGINNING OF TIME STEP IN C
C     TFM=   LIQUID BULK TEMP. IN C
C     TSM=   SAT'N TEMP. CORRESPONDING TO P IN C
C     HFGM=  LATENT HEAT AT P IN KJ/KG
C     VFM=   SAT'T LIQUID SPECIFIC VOL. IN M**3/KG
C     VGM=   SAT'T VAPOUR SPECIFIC VOL. IN M**3/KG
C     TINTE= FLUID TEMP. IN F
C     QOAE=  HEAT INPUT PER UNIT AREA IN BTU/(HR*SQ.FT)
C     VHTCO= VAPOUR HEAT TRANSFER COEFF. IN BTU/(HR*SQ.FT*F)
C     LHTCO= LIQUID HEAT TRANSFER COEFF. IN BTU/(HR*SQ.FT*F)
C     HI(I)= INTERNAL HEAT TRANFER COEFF. IN J/(SQ.M*SEC*C)
C     HTCOMX=MAXIMUM HEAT TRANSFER COEFF. IN W/SQ.M*C
C     LC=CHARACTERISTIC LENGTH IN M
C     L=LATENT HEAT OF EVAPORATION IN BTU/LB
C     BETAF=THERMAL COEFF. OF EXPANSION (1/R)
C     MUE=DYNAMIC VISCOSITY
C     SIGE=SURFACE TENSION OF PROPANE (DYNES/CM)
C
      IMPLICIT REAL (A-H,O-Z)
      REAL L,L3,LC,LC3,LHTCO,MUE,MUM
      COMMON / INDEX / NBEG,NTOT,NANG
      COMMON / MESH / THETA,DELTH,DTHETA,R,DR,DELTR
      DIMENSION HI(21),TW(21,15),QOA(21),THETA(21),DELTH(21),R(15),
     &DR(15),DELTR(15)
C     INITIALIZE HI(I)
      IF(TIME.GT.0.0) GO TO 5
      DO 6 I=2,NANG
        HI(I)=0.C
 6        CONTINUE
C
C       VAPOUR HEAT TRANSFER COEFF. IS CONSTANT @ 22.0 W/SQM K
C
 5      IF(IOPEN.EQ.1)HTCOMX=13000.
        IF(IOPEN.EQ.0)HTCOMX=13000.
        DENSFM=1.0/VFM
        VHTCO=22.0*0.1761
C       CONVERT PRIMATIVE PROPS. TO ENGLISH UNITS
        PE=PM*14.5
        TCRT=714.24
C       CHAR. NO. IS A/P
C        LC=0.0531976279
C       CHAR. NO. IS WETTED SURFACE LENGTH IN FT.
        LC=WETSRF*3.2808
        LC3=LC*LC*LC
        VFE=VFM*16.01846
        VGE=VGM*16.01846
        L=HFGM*0.4299
        L3=L*L*L
        VGMF=VGE-VFE
        VGOF=VGE/VFE
        G=417.312E+06
        TFE=TFM*(9./5.)-460.
C
C       RAYLEIGH NO. CRITERION FOR FREE CONVECTION
C
        ARA=1.0E+04
        BRA=1.0E+09
        CRA=1.0E+13
        DO 1 I=ILEVEL,NANG
          TWINS=TW(I,NBEG)*(9./5.)-460.
          TINTE=(TWINS+TFE)/2.
C       USE SAT'D LIQUID DENS. & FILM TEMP. FOR COND. & VISCOS.
          TINTM=(TINTE+460.)*5./9.
          CPM=CPSL(TINTM)
          CPE=CPM*5.415873E-03
          CONDM=CONDUC(TINTM,DENSFM)
          CONDE=CONDM*577.8E-06
C       CONDUCTION HEAT TRANSFER FOR FIRST TIME STEP
          IF(TIME.EQ.0.0) GO TO 3
          BETAF=0.06284/(TCRT-(TFE+460.))**0.641
          MUM=VISCOS(TINTM,DENSFM)
          MUE=MUM*241.92E-06
          RA=G*BETAF*(TWINS-TFE)*LC3*CPE/(MUE*CONDE*VFE*VFE)
C
C       EVALUATE THE LIQUID HEAT TRANSFER COEFFICIENT
C
          IF(RA.LE.CRA)THEN DO
C
C         LIQUID IS SUBCOOLED
C
            CDOL=CONDE/LC
C
C         ASSUMING A ISOTHERMAL VERTICAL PLATE
C
            IF(RA.GE.ARA.AND.RA.LT.BRA) THEN DO
              LHTCO=0.59*CDOL*(RA**0.25)
              ELSE DO
              IF(RA.GE.BRA.AND.RA.LE.CRA) THEN DO
```

```
               LHTCO=0.10*CDOL*(RA**0.3333333)
               END IF
             END IF
          ELSE DO
C
C      LIQUID IS TWO-PHASE
C
           IF(PE.LE.168.0) THEN DO
             QOAE=QOA(I)*0.317
             SIGM=SIGMA(TINTM)
             SIGE=SIGM*68.5221E-06
             VGOF2=VGOF*VGOF
             IF(QOAE.LE.0.0) GO TO 4
             Z=(CPE*CPE*SIGE*(TINTE+460.)*QOAE*VGOF2*VGMF/(CONDE*
     &         L3))**0.416667
             LHTCO=(1.86276*CPE*QOAE*VGOF/L)/Z
             ELSE DO
             LHTCO=(15.+(0.0642E-06*(PE**3.347)))*((ABS(TWINS-TFE))
     &               **1.55)
             END IF
           END IF
         GO TO 4
C      FIRST TIME STEP IS CONDUCTION WITH B. L. OF 1/8 IN.
 3       LHTCO=CONDE*96.
 4       CONTINUE
C      TO KEEP LHTCO FROM OSCILLATING FROM CONVECTION TO BOILING
C      THE MAX. VALUE WILL BE CONSIDERED TO PREVAIL
         HTOLD=HI(I)*0.1761
         HI(I)=LHTCO
         IF(HTOLD.GT.HI(I)) HI(I)=HTOLD
 1       CONTINUE
      DO 2 I=2,NANG
        IF(I.LT.ILEVEL) THEN DO
          HI(I)=VHTCO/0.1761
          ELSE DO
          HI(I)=HI(I)/0.1761
C
C       SET THE MAXIMUM HEAT TRANSFER COEFFICIENT
C
          IF(HI(I).GE.HTCOMX) HI(I)=HTCOMX
          END IF
 2      CONTINUE
      RETURN
      END
C****************************************
      SUBROUTINE ILIQ (VOLG,RIS,XLTH,DTHETA,ILEVEL,WETSRF,TAUB,
     &AREA)
C****************************************
C
C     THIS SUBROUTINE EVALUATES THE LIQUD LEVEL, THE INTERFACIAL
C     AREA AND THE MEAN BUBBLE TRAVEL LENGTH
C
      IMPLICIT REAL*8 (A-H,O-Z)
      REAL*4 X,VOLG,RIS,DTHETA,TAUB,TAUC,XLTH,AREA,WETSRF
      INTEGER ITMAX,ITER,I,ILEVEL
      PI=3.141592654
C-----------------------------------------------------
C     RIS=INSIDE RADIUS OF TANK IN METRES
C     VOLG=VOLUME OF VAPOUR
C     VOLPL=VOLUME OF VAPOUR / UNIT LENGTH IN M**3/m
```

```
C     EPS=TOLERANCE
C     AREA=INTERFACIAL AREA IN M
C------------------------------------------------------------
      EPS=1.0E-05
      ITMAX=100
C     FIRST GUESS FOR Y
      ANGLE=3.14
      ITER=0
      VOLPL=VOLG/XLTH
      DO 1 I=1,ITMAX
        FY=RIS**2*(ANGLE-DSIN(ANGLE))/2.0-VOLPL
        FPY=RIS**2*(1.0-DCOS(ANGLE))/2.0
C    TO ALLOW FOR Y=0.0*, 360*
        IF(FPY.EQ.0.0)FPY=1.0E10
        YKP1=ANGLE-FY/FPY
        IF(DABS(YKP1-ANGLE).GT.EPS) GO TO 2
        GO TO 3
 2      ANGLE=YKP1
        ITER =ITER+1
 1      CONTINUE
 3    CONTINUE
      ANGLE=YKP1
      X=ANGLE/(2.0*DTHETA)+2.5
      ILEVEL=IFIX(X)
C     DETERMINE THE MEAN PATH LENGTHS OF BUBBLES & DROPS
C      TAUC=VOLPL/(2.0*RIS)
      TAUB=(PI*RIS**2.0-VOLPL)/(2.0*RIS)
C
C     WETTED SURFACE LENGTH IN M
C
      WETSRF=(2.0*PI-ANGLE)*RIS/2.0
C     FOR ANGLE WHICH IS .LE. THAN 180
      IF(ANGLE.LE.PI) THEN DO
        ANGLEH=ANGLE/2.0
        ELSE DO
C     FOR ANGLE WHICH IS .GT. 180
        ANGLEH=(2.0*PI-ANGLE)/2.0
        END IF
      AREA=2.0*DCOS(ANGLEH)*XLTH
      RETURN
      END
C
C-----------------------------------------------------
      SUBROUTINE HTINFD(VOLG,XLTH,TGB,TFB,ILEVEL,TN,QV,QL,QOA)
C-----------------------------------------------------
C
C     THIS SUBROUTINE DETERMINES THE HEAT INPUT
C     TO THE CONTROL VOLUMES FROM THE SHELL
C                   IN KJ/S
C
      IMPLICIT REAL (A-H,O-Z)
      DIMENSION DR(15),DELTR(15),R(15),THETAD(21),THETA(21)
     &,DELTH(21),AE(21,15),AW(21,15),AN(21,15),AS(21,15),APO(21,15),
     &TN(21,15),QIN(21),HI(21),QOA(21)
      COMMON / COEFFB / AE,AW,AN,AS,APO,QIN,HI
      COMMON / MESH / THETA,DELTH,DTHETA,R,DR,DELTR
      COMMON / INDEX / NBEG,NTOT,NANG
C     HEAT FLUX IN EACH ELEMENT
      QV=0.0
      QL=0.0
      DO 1 I=2,NANG
        IF(I.LT.ILEVEL) TINF=TGB
        IF(I.EQ.ILEVEL) TINF=(TGB+TFB)/2.
        IF(I.GT.ILEVEL) TINF=TFB
        HEAT=HI(I)*R(NBEG)*DELTH(I)*(TN(I,NBEG)-TINF)
        QOA(I)=HEAT/(R(NBEG)*DELTH(I))
C       CAN'T HAVE TOO MUCH HEAT INTO THE VAPOUR
        IF(I.LT.ILEVEL) THEN DO
          QV=QV+HEAT
          ELSE DO
          QL=QL+HEAT
          END IF
 1      CONTINUE
C     MUST DOUBLE THE HEAT INPUT SINCE ONLY 1/2 HAS BEEN CONSIDERED
      QV=2.0*QV/1000.
      QL=2.0*QL/1000.
C     MULT BY LENGTH TO ACCOUNT FOR FULL LENGTH
      QV=QV*XLTH
      QL=QL*XLTH
      RETURN
      END
C
C*****************************************************
C*****************************************************
C        PROPERTIES AND DERIVATIVES OF PROPANE
C*****************************************************
C*****************************************************
C
C-----------------------------------------
      SUBROUTINE SUBP
C-----------------------------------------
C
C     THE PROPERTIES OF SUBCOOLED PROPANE
C
      IMPLICIT REAL (A-H,O-Z)
      COMMON / SUBPP / PCE,HCE,DENSCE,TCE,NDPCI,NDPCJ
      DIMENSION DENSCE(14),PCE(14,14),TCE(14,14),HCE(14,14)
      DATA NDPCI,NDPCJ / 14,14 /
C
C     DENSCE: DENSITY (KG/CU.M)
C
      DATA (DENSCE(K),K=1,NDPCI) / 572.,560.,547.,535.,520.,505.,490.,
     Z473.,455.,435.,412.,385.,346.,317. /
C
C     TCE: TEMPERATURE (K)
C
      DATA (TCE(1,K),K=1,NDPCJ) / 238.5,239.0 ,239.75,240.,240.2,240.4,
     Z240.6,240.8,241.0,241.4,241.8,242.2,242.6,243.0 /
      DATA (TCE(2,K),K=1,NDPCJ) / 248.463,248.75,249.0,249.25,249.5,
     Z249.75,250.0,250.2,250.6,251.0,251.4,251.8,252.4,253.2 /
      DATA (TCE(3,K),K=1,NDPCJ) / 258.914,259.1,259.2,259.4,259.6,
     Z259.8,260.0,260.5,261.0,261.5,262.0,262.5,263.0,264.0 /
      DATA (TCE(4,K),K=1,NDPCJ) / 268.193,268.50,268.75,269.00,269.25,
     Z269.50,269.75,270.00,270.50,271.0,271.50,272.00,272.50,273.5 /
      DATA (TCE(5,K),K=1,NDPCJ) / 279.261,279.75,280.0,280.3,280.6,
     Z280.9,281.2,281.50,281.8,282.1,282.4,283.0,283.9,285.1 /
      DATA (TCE(6,K),K=1,NDPCJ) / 289.714,290.0,290.35,290.7,291.05,
     Z291.40,291.75,292.1,292.45,292.8,293.15,293.85,294.9,295.6 /
      DATA (TCE(7,K),K=1,NDPCJ) / 299.53,300.0,300.35,300.7,301.05,
     Z301.40,301.75,302.1,302.45,302.8,303.15,303.5,304.55,305.6 /
```

```
      DATA (TCE(8,K),K=1,NDPCJ) / 309.86,310.35,310.7,311.05,311.4,
     Z311.75,312.1,312.45,312.8,313.5,314.2,314.9,315.6,316.3 /
      DATA (TCE(9,K),K=1,NDPCJ) / 319.862,320.35,320.7,321.05,321.4,
     Z321.75,322.1,322.45,323.15,323.85,324.55,325.25,325.6,326.3 /
      DATA (TCE(10,K),K=1,NDPCJ) / 329.836,330.35,330.7,331.05,331.4,
     Z331.75,332.1,332.45,332.8,333.15,333.85,334.55,335.25,336.3 /
      DATA (TCE(11,K),K=1,NDPCJ) / 339.82,340.35,340.7,341.05,341.4,
     Z341.75,342.1,342.45,342.8,343.15,343.85,344.55,345.25,345.95 /
      DATA (TCE(12,K),K=1,NDPCJ) / 349.535,350.0,350.3,350.6,350.9,
     Z351.2,351.5,351.8,352.1,352.4,352.7,353.0,353.90,354.8 /
      DATA (TCE(13,K),K=1,NDPCJ) / 359.86,360.25,360.5,360.75,361.0,
     Z361.25,361.5,361.75,362.0,362.25,362.5,362.75,363.0,363.25 /
      DATA (TCE(14,K),K=1,NDPCJ) / 364.89,365.1,365.2,365.25,365.3,
     Z365.4,365.5,365.6,365.7,365.8,365.9,366.0,366.1,366.2 /
C
C     HCE: ENTHALPY (KJ/KG)
C
      DATA (HCE(1,K),K=1,NDPCJ) / 438.61,440.16,442.49,443.26,443.88.
     Z444.50,445.12,445.74,446.37,447.61,448.86,450.10,451.34,452.59 /
      DATA (HCE(2,K),K=1,NDPCJ) / 461.917,462.79,463.55,464.31,465.07,
     Z465.83,466.59,467.20,468.41,469.63,470.84,472.06,473.89,476.33 /
      DATA (HCE(3,K),K=1,NDPCJ) / 487.041,487.60,487.89,488.48,489.08,
     Z489.67,490.27,491.76,493.24,494.73,496.21,497.70,499.19,502.17 /
      DATA (HCE(4,K),K=1,NDPCJ) / 509.917,510.81,511.54,512.27,512.99,
     Z513.72,514.45,515.18,516.64,518.10,519.56,521.02,522.48,525.40 /
      DATA (HCE(5,K),K=1,NDPCJ) / 537.968,539.36,540.07,540.92,541.77,
     Z542.63,543.48,544.34,545.19,546.05,546.91,548.62,551.19,554.62 /
      DATA (HCE(6,K),K=1,NDPCJ) / 565.292,566.09,567.06,568.04,569.02,
     Z569.99,570.97,571.95,572.93,573.90,574.88,576.84,579.77,581.73 /
      DATA (HCE(7,K),K=1,NDPCJ) / 591.807,593.09,594.05,595.00,595.96,
     Z596.92,597.88,598.84,599.79,600.75,601.71,602.67,605.56,608.44 /
      DATA (HCE(8,K),K=1,NDPCJ) / 620.768,622.08,623.02,623.95,624.89,
     Z625.83,626.78,627.71,628.65,630.54,632.42,634.31,636.19,638.09 /
      DATA (HCE(9,K),K=1,NDPCJ) / 650.057,651.34,652.27,653.18,654.11,
     Z655.03,655.95,656.88,658.73,660.58,662.44,664.30,665.22,667.08 /
      DATA (HCE(10,K),K=1,NDPCJ) / 680.838,682.17,683.07,683.98,684.89,
     Z685.81,686.71,687.62,688.53,689.44,691.27,693.09,694.92,697.67 /
      DATA (HCE(11,K),K=1,NDPCJ) / 713.754,715.11,716.00,716.89,717.79,
     Z718.69,719.58,720.48,721.37,722.27,724.07,725.87,727.67,729.48 /
      DATA (HCE(12,K),K=1,NDPCJ) / 748.732,749.91,750.66,751.42,752.17,
     Z752.93,753.68,754.44,755.20,755.96,756.72,757.47,759.76,762.04 /
      DATA (HCE(13,K),K=1,NDPCJ) / 791.838,792.80,793.42,794.05,794.66,
     Z795.29,795.91,796.53,797.16,797.78,798.40,799.03,799.66,800.29 /
      DATA (HCE(14,K),K=1,NDPCJ) / 818.29,818.81,819.07,819.19,819.31,
     Z819.57,819.82,820.07,820.32,820.57,820.82,821.08,821.33,821.58 /
C
C     PCE: PRESSURE (BAR)
C
      DATA (PCE(1,K),K=1,NDPCJ) / 1.665,6.164,12.912,15.16,16.960,
     Z18.758,20.556,22.356,24.154,27.751,31.348,34.945,38.541,42.135 /
      DATA (PCE(2,K),K=1,NDPCJ) / 2.0603,4.430,6.498,8.565,10.632,
     Z12.699,14.768,15.422,19.728,23.035,26.341,29.647,34.605,41.214 /
      DATA (PCE(3,K),K=1,NDPCJ) / 2.9988,4.395,5.151,6.662,8.174,9.685,
     Z11.197,14.975,18.753,22.530,26.306,30.082,33.858,41.408 /
      DATA (PCE(4,K),K=1,NDPCJ) / 4.073,6.211,7.952,9.692,11.432,
     Z13.171,14.910,16.650,20.129,23.607,27.085,30.562,34.038,40.990 /
      DATA (PCE(5,K),K=1,NDPCJ) / 5.7041,8.773,10.344,12.227,14.112,
     Z15.996,17.881,19.766,21.647,23.532,25.416,29.183,34.832,42.361 /
      DATA (PCE(6,K),K=1,NDPCJ) / 7.6469,9.263,11.248,13.234,15.218,
     Z17.203,19.190,2..173,23.158,25.141,27.125,31.093,37.043,41.008 /
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       DATA (PCE(7,K),K=1,NDPCJ) / 9.8793,12.286,14.076,15.870,17.661,
      Z19.452,21.245,23.036,24.827,26.618,28.410,30.200,35.570,40.940 /
       DATA (PCE(8,K),K=1,NDPCJ) / 12.71,14.941,16.536,18.129,19.724,
      Z21.142,22.911,24.506,26.099,29.286,32.473,35.658,38.843,42.028 /
       DATA (PCE(9,K),K=1,NDPCJ) / 15.9772,17.936,19.342,20.747,22.154,
      Z23.561,24.966,26.372,29.183,31.995,34.805,37.616,39.021,41.831 /
       DATA (PCE(10,K),K=1,NDPCJ) / 19.8078,21.598,22.818,24.037,25.257,
      Z26.477,27.697,28.917,30.136,31.356,33.796,36.235,38.675,42.335 /
       DATA (PCE(11.K),K=1,NDPCJ) / 24.2744,25.835,26.867,27.898,28.931,
      Z29.963,30.994,32.027,33.059,34.092,36.157,38.223,40.290,42.356 /
       DATA (PCE(12,K),K=1,NDPCJ) / 29.296,30.418,31.141,31.864,32.587,
      Z33.311,34.035,34.759,35.483,36.207,36.932,37.657,39.832,42.008 /
       DATA (PCE(13,K),K=1,NDPCJ) / 35.4608,36.156,36.605,37.053,37.503,
      Z37.952,38.401,38.851,39.301,39.752,40.202,40.653,41.104,41.556 /
       DATA (PCE(14,K),K=1,NDPCJ) / 38.8232,39.124,39.268,39.341,39.413,
      Z39.557,39.702,39.846,39.991,40.136,40.281,40.426,40.571,40.717 /
       RETURN
       END
C-----------------------------
       SUBROUTINE SUPP
C-----------------------------
C
C      THE PROPERTIES OF SUPERHEATED PROPANE
C
       IMPLICIT REAL (A-H,O-Z)
       COMMON / SUPPP / PE,HHE,DENSHE,THE,NDPHI,NDPHJ
       DIMENSION DENSHE(17),PE(17,24),THE(17,24),HHE(17,24)
       DATA NDPHI,NDPHJ / 17,24 /
C
C      DENSH: DENSITY (KG/CU.M)
C
       DATA (DENSHE(K),K=1,NDPHI) / 100.,76.9,58.7,45.8,35.8,27.7,21.5,
      Z16.5,12.0,9.35,7.5,6.9,5.9,4.93,4.2,3.35,3.1 /
C
C      THE: TEMPERATURE (F)
C
       DATA (THE(17,K),K=1,NDPHJ) / 238.,360.,390.,450.,480.,540.,570.,
      Z630.,660.,720.,750.,780.,840.,870.,930.,960.,1020.,1050.,1110.,
      Z1200.,1260.,1320.,1410.,1620. /
       DATA (THE(16,K),K=1,NDPHJ) / 240.,260.,280.,300.,320.,340.,360.,
      Z380.,400.,420.,440.,460.,480.,500.,520.,540.,560.,580.,600.,
      Z640.,700.,990.,1260.,1590. /
       DATA (THE(15,K),K=1,NDPHJ) / 245.368,270.,290.,310.,350.,390.,
      Z430.,470.,510.,550.,590.,630.,670.,710.,750.,790.,830.,870.,950.,
      Z1090.,1220.,1370.,1490.,1610. /
       DATA (THE(14,K),K=1,NDPHJ) / 250.,270.,290.,310.,330.,350.,
      Z370.,390.,410.,430.,450.,470.,490.,510.,530.,550.,570.,590.,610.,
      Z630.,700.,1090.,1360.,1600. /
       DATA (THE(13,K),K=1,NDPHJ) / 255.145,290.,310.,330.,350.,390.,
      Z430.,470.,510.,550.,590.,630.,590.,730.,770.,810.,850.,890.,930.,
      Z1010.,1130.,1270.,1420.,1600. /
       DATA (THE(12,K),K=1,NDPHJ) / 259.772,270.,280.,300.,320.,340.,
      Z360.,380.,400.,420.,440.,460.,480.,500.,520.,540.,560.,580.,600.,
      Z620.,700.,1010.,1310.,1610. /
       DATA (THE(11.K),K=1,NDPHJ) / 262.486,300.,320.,340.,360.,400.,
      Z440.,480.,520.,560.,600.,640.,680.,720.,760.,800.,840.,880.,980.,
      Z1080.,1190.,1340.,1490.,1610. /
       DATA (THE(10,K),K=1,NDPHJ) / 269.548,280.,290.,310.,330.,360.,
      Z370 ,390.,410.,430.,450.,470.,490.,510.,530.,550.,570.,590.,610.,
      Z630.,700.,1020.,1320.,1590. /
      DATA (THE(9,K),K=1,NDPHJ) / 278.105,310.,330.,350.,390.,430.,470.,
     Z510.,550.,590.,630.,670.,710.,750.,790.,830.,870.,910.,970.,
     Z1050.,1150.,1310.,1460.,1610. /
      DATA (THE(8,K),K=1,NDPHJ) / 289.531,
     Z300.,310.,320.,330.,340.,350.,370.,390.,410.,430.,450.,470.,490.,
     Z510.,530.,550.,570.,590.,610.,700.,815.,1090.,1340. /
      DATA (THE(7,K),K=1,NDPHJ) / 299.565,310.,320.,330.,340.,350.,360.,
     Z380.,400.,420.,440.,460.,480.,500.,520.,540.,560.,580.,600.,620.,
     Z700.,800.,900.,1040. /
      DATA (THE(6,K),K=1,NDPHJ) / 309.531,320.,330.,340.,350.,360.,370.,
     Z380.,400.,420.,440.,460.,480.,500.,520.,540.,560.,580.,600.,620.,
     Z640.,700.,760.,820. /
      DATA (THE(5,K),K=1,NDPHJ) /319.8623,
     Z330.,340.,350.,360.,370.,380.,390.,400.,410.,420.,
     Z440.,460.,480.,500.,520.,540.,560.,580.,600.,620.,630.,640.,650. /
      DATA (THE(4,K),K=1,NDPHJ) / 329.874,335.,340.,345.,
     Z350.,360.,370.,380.,390.,400.,410.,420.,
     Z430.,440.,450.,460.,470.,480.,490.,500.,510.,520.,530.,540. /
      DATA (THE(3,K),K=1,NDPHJ) / 339.792,345.,350.,355.,360.,365.,
     Z370.,375.,380.,385.,390.,395.,400.,405.,410.,415.,420.,425.,
     Z430.,435.,440.,445.,450.,460. /
      DATA (THE(2,K),K=1,NDPHJ) / 349.94,360.,362.,364.,366.,368.,
     Z370.,372.,374.,376.,378.,380.,382.,384.,386.,388.,390.,392.,
     Z394.,396.,398.,400.,404.,408. /
      DATA (THE(1,K),K=1,NDPHJ) / 358.5314,360.,361.,362.,363.,364.,
     Z365.,366.,367.,368.,369.,370.,371.,372.,373.,374.,375.,376.,
     Z377.,378.,379.,380.,381.,383. /
C
C     HHE: ENTHALPY (KJ/KG)
C
      DATA (HHE(17,K),K=1,NDPHJ) / 857.85,1064.77,1125.63,1259.07,
     Z1331.48,1487.03,1569.91,1745.18,1837.31,2029.93,2130.17,
     Z2232.89,2445.32,2554.84,2780.03,2895.53,3131.98,3252.79,
     Z3499.20,3879.98,4140.66,4406.37,4813.59,5799.33 /
      DATA (HHE(16,K),K=1,NDPHJ) / 860.25,890.11,
     Z921.54,954.65,989.48,1026.08,
     Z1064.45,1104.59,1146.49,
     Z1190.14,1235.50,1282.55,1331.23,
     Z1381.53,1433.40,1486.81,
     Z1541.71,1598.05,1655.80,1775.38,
     Z1964.36,3012.76,4140.58,5655.68  /
      DATA (HHE(15,K),K=1,NDPHJ) / 866.446,904.15,936.53,970.59,
     Z1043.93,1124.33,1211.69,1305.84,1406.50,1513.35,1626.07,
     Z1744.35,1867.88,1996.35,2129.48,2267.00,2408.66,2554.25,
     Z2856.31,3415.91,3965.94,4631.08,5183.36,5751.18 /
      DATA (HHE(14,K),K=1,NDPHJ) / 871.97,902.90,
     Z935.38,969.52,1005.38,1042.99,
     Z1082.35,1123.48,1166.34,
     Z1210.92,1257.20,1305.13,1354.70,
     Z1405.84,1458.54,1512.74,
     Z1568.41,1625.50,1683.99,1743.82,1963.30,3415.64,4585.63,
     Z5703.21 /
      DATA (HHE(13,K),K=1,NDPHJ) / 877.97,933.85,968.10,1004.05,
     Z1041.74,1122.36,1209.90,1304.20,1404.98,1511.94,1624.75,
     Z1743.12,1930.39,2051.29,2196.70,2336.39,2480.10,2627.61,
     Z2778.69,3090.90,3581.68,4183.80,4858.82,5703.08 /
      DATA (HHE(12,K),K=1,NDPHJ) / 883.36,899.49,915.68,
     Z949.24,984.45,1021.35,1059.99,
     Z1100.37,1142.48,1186.31,
     Z1231.84,1279.03,1327.86,1378.29,
     Z1430.28,1483.81,1538.80,
     Z1595.24,1653.09,1712.30,1962.04,3090.50,4360.74,5750.84 /
      DATA (HHE(11,K),K=1,NDPHJ) / 886.477,948.33,983.60,1020.56,
     Z1059.24,1141.81,1231.23,1327.30,1429.77,1538.32,1652.64,
     Z1772.42,1897.35,2027.15,2161.53,2300.22,2442.99,2589.60,
     Z2971.66,3373.56,3835.66,4494.91,5182.79,5750.77 /
      DATA (HHE(10,K),K=1,NDPHJ) / 894.517,911.60,928.37,
     Z963.05,999.34,1037.32,
     Z1077.01,1118.41,1161.52,1206.32,
     Z1252.80,1300.92,1350.66,
     Z1401.96,1454.80,1509.13,1564.93,
     Z1622.14,1680.75,1740.69,1960.50,3129.47,4404.84,5654.91 /
      DATA (HHE(9,K),K=1,NDPHJ) / 903.977,959.17,995.74,1033.94,
     Z1115.40,1203.60,1298.43,1399.67,1507.02,1620.17,1738.85,
     Z1862.72,1991.52,2124.94,2262.73,2404.64,2550.47,2699.98,
     Z2930.69,3249.46,3664.02,4359.63,5042.49,5750.40 /
      DATA (HHE(8,K),K=1,NDPHJ) / 916.067,934.53,
     Z952.54,970.89,989.62,1008.72,1028.23,1068.46,
     Z1110.33,1153.85,1199.02,1245.83,
     Z1294.26,1344.27,1395.82,
     Z1448.90,1503.46,1559.47,1616.88,
     Z1675.67,1956.15,2348.72,3411.85,4493.23 /
      DATA (HHE(7,K),K=1,NDPHJ) / 926.02,
     Z945.14,963.82,982.82,1002.18,1021.91,1042.02,
     Z1083.43,1126.44,1171.07,1217.31,
     Z1265.16,1314.57,1365.55,
     Z1418.05,1472.04,1527.48,1584.33,
     Z1642.57,1702.14,1953.19,2293.05,2658.23,3205.85 /
      DATA (HHE(6,K),K=1,NDPHJ) / 935.08,
     Z955.02,974.42,994.11,1014.13,1034.50,1055.25,1076.36,
     Z1119.77,1164.74,1211.30,
     Z1259.42,1309.09,1360.30,1413.01,
     Z1467.21,1522.83,1579.86,
     Z1638.26,1698.00,1759.05,1949.61,2150.66,2361.28 /
      DATA (HHE(5,K),K=1,NDPHJ) /
     Z943.323,963.45,983.62,1004.05,1024.78,1045.85,1067.26,
     Z1089.04,1111.18,1133.70,1156.60,1203.56,
     Z1252.04,1302.04,1353.55,1406.55,
     Z1461.00,1516.87,1574.14,
     Z1632.76,1692.70,1723.17,1753.95,1785.03 /
      DATA (HHE(4,K),K=1,NDPHJ) / 949.795,960.32,970.76,981.23,
     Z991.74,1012.94,1034.42,1056.20,1078.31,
     Z1100.76,1123.57,1146.74,1170.28,1194.19,1218.46,1243.11,
     Z1268.12,1293.51,1319.27,1345.39,1371.88,1398.73,1425.94,
     Z1453.51 /
      DATA (HHE(3,K),K=1,NDPHJ) / 953.991,965.18,976.09,987.00,
     Z997.95,1008.94,1019.98,1031.08,1042.24,1053.48,1064.78,
     Z1076.17,1087.63,1099.18,1110.81,1122.52,1134.31,1146.20,
     Z1158.16,1170.22,1182.37,1194.61,1206.93,1231.85 /
      DATA (HHE(2,K),K=1,NDPHJ) / 954.515,977.35,981.93,986.5C,
     Z991.07,995.64,1000.22,1004.79,1009.39,1013.98,1018.58,
     Z1023.18,1027.79,1032.41,1037.04,1041.68,1046.33,1050.98,
     Z1055.65,1060.33,1065.02,1069.71,1079.15,1088.62 /
      DATA (HHE(1,K),K=1,NDPHJ) / 948.854,952.24,954.69,957.12,
     Z959.55,961.96,964.38,966.79,969.19,971.58,973.98,976.38,
     Z978.77,981.14,983.54,985.92,988.31,990.69,993.07,995.46,
     Z997.84,1000.22,1002.61,1007.38 /
C
C     PE: PRESSURE (BAR)
C
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 DATA (PE(17,K),K=1,NDPHJ) / 1.327,2.062,2.240,2.596,2.774,
Z3.129,3.306,3.660,3.836,4.190,4.366,4.543,4.896,5.072,
Z5.425,5.601,5.954,6.130,6.483,7.011,7.364,7.716,8.244,
Z9.477 /
 DATA (PE(16,K),K=1,NDPHJ) / 1.442,1.574,
Z1.706,1.836,1.966,2.096,2.225,
Z2.354,2.482,2.611,2.739,2.867,
Z2.995,3.123,3.251,3.379,3.507,
Z3.634,3.762,4.017,4.399,6.243,7.958,10.052 /
 DATA (PE(15,K),K=1,NDPHJ) / 1.8306,2.037,2.203,2.367,2.694,
Z3.019,3.343,3.665,3.987,4.308,4.629,4.950,5.270,5.590,
Z5.910,6.230,6.549,6.869,7.507,8.624,9.660,10.855,11.811,
Z12.766 /
 DATA (PE(14,K),K=1,NDPHJ) / 2.174,2.372,2.568,
Z2.762,2.955,3.148,3.340,3.531,
Z3.722,3.912,4.102,4.292,4.482,
Z4.671,4.860,5.049,5.238,5.427,
Z5.616,5.804,6.463,10.123,12.652,14.897 /
 DATA (PE(13,K),K=1,NDPHJ) / 2.6295,3.044,3.279,3.513,3.745,
Z4.206,4.664,5.121,5.576,6.030,6.484,6.936,7.614,8.065,
Z8.516,8.967,9.417,9.868,10.318,11.217,12.565,14.136,15.819,
Z17.837 /
 DATA (PE(12,K),K=1,NDPHJ) / 3.0754,3.246,3.386,
Z3.665,3.941,4.215,4.488,4.759,
Z5.030,5.299,5.568,5.836,6.104,
Z6.371,6.638,6.904,7.170,7.436,
Z7.702,7.967,9.027,13.117,17.063,21.003 /
 DATA (PE(11,K),K=1,NDPHJ) / 3.3797,3.962,4.264,4.563,4.861,
Z5.452,6.038,6.622,7.204,7.784,8.363,8.941,9.518,10.094,10.670,
Z11.245,11.819,12.393,13.828,15.260,16.834,18.980,21.123,
Z22.837 /
 DATA (PE(10,K),K=1,NDPHJ) / 4.2379,4.469,4.664,
Z5.049,5.429,5.806,6.180,6.553,
Z6.923,7.292,7.660,8.027,8.393,
Z8.758,9.123,9.487,9.850,
Z10.213,10.575,10.937,12.202,17.953,23.321,28.144 /
 DATA (PE(9,K),K=1,NDPHJ) / 5.507,6.334,6.832,7.326,
Z8.299,9.262,10.216,11.164,12.108,13.048,13.985,
Z14.920,15.853,16.784,17.714,18.643,19.571,20.498,
Z21.887,23.737,26.046,29.735,33.189,36.640 /
 DATA (PE(8,K),K=1,NDPHJ) / 7.5927,8.008,
Z8.370,8.727,9.082,9.433,9.782,10.473,11.158,
Z11.837,12.512,13.183,13.850,
Z14.515,15.177,15.837,16.495,
Z17.152,17.807,18.461,21.389,25.109,33.944,41.935 /
 DATA (PE(7,K),K=1,NDPHJ) / 9.8716,10.420,
Z10.907,11.387,11.862,12.333,12.799,13.723,
Z14.635,15.539,16.436,17.327,
Z18.213,19.094,19.972,20.846,
Z21.718,22.586,23.452,24.317,
Z27.756,32.026,36.274,42.195 /
 DATA (PE(6,K),K=1,NDPHJ) / 12.5937,
Z13.321,13.971,14.611,15.243,15.868,16.487,17.101,
Z18.316,19.517,20.705,21.884,
Z23.054,24.217,25.374,26.525,
Z27.672,28.814,29.952,31.087,
Z32.219,35.599,38.960,42.305 /
 DATA (PE(5,K),K=1,NDPHJ) / 15.956,
Z16.903,17.779,18.640,19.489,20.327,21.156,21.978,22.793,
Z23.602,24.405,25.997,27.573,29.135,
     Z30.685,32.224,33.755,35.279,
     Z36.795,38.305,39.809,40.560,41.309,
     Z42.057 /
      DATA (PE(4,K),K=1,NDPHJ) / 19.801,20.461,21.058,21.648,
     Z22.232,23.382,24.514,25.631,26.735,27.828,
     Z28.911,29.985,31.051,32.109,33.162,34.208,35.248,36.284,
     Z37.315,38.342,39.365,40.384,41.399,42.411 /
      DATA (PE(3,K),K=1,NDPHJ) / 24.237,25.129,25.936,26.732,
     Z27.519,28.297,29.067,29.831,30.589,31.342,32.090,32.833,
     Z33.571,34.306,35.037,35.764,36.488,37.209,37.926,38.641,
     Z39.353,40.063,40.770,42.178 /
      DATA (PE(2,K),K=1,NDPHJ) / 29.497,31.817,32.262,32.705,
     Z33.146,33.584,34.020,34.454,34.887,35.317,35.746,36.173,
     Z36.599,37.023,37.446,37.867,38.288,38.707,39.124,39.541,
     Z39.956,40.370,41.196,42.017 /
      DATA (PE(1,K),K=1,NDPHJ) / 34.601,35.083,35.404,35.723,36.040,
     Z36.356,36.670,36.983,37.294,37.605,37.914,38.223,38.530,38.836,
     Z39.142,39.447,39.751,40.054,40.356,40.658,40.959,41.259,41.559,
     Z42.156 /
      RETURN
      END
C------------------------------
      SUBROUTINE SATP
C------------------------------
C
C     NDPP-  NO. OF DATA POINT PROPERTIES
C
      IMPLICIT REAL (A-H,O-Z)
      COMMON / SATPP / TS,PS,HFS,HGS,VFS,VGS,NDPP
      DIMENSION PS(27),TS(27),VGS(27),VFS(27),HGS(27),HFS(27),
     ZHFG(27),VFG(27)
      DIMENSION RHOGS(27),RHOFS(27)
      DATA NDPP / 27 /
C     PRINT,'************************************'
C     PRINT,'       SATURATED PROPERTIES          '
C     PRINT,'************************************'
C
C     PS:  PRESSURE(BAR)
C
      DATA (PS(K),K=1,NDPP)/1.48,1.805,2.182,2.616,3.112,
     Z3.675,4.312,5.027,5.828,6.718,7.706,8.797,9.997,11.313,
     Z12.752,14.321,16.026,17.875,19.876,22.035,24.361,26.865,
     Z29.555,32.444,35.55,38.9,42.38/
C
C     TS:  TEMPERATURE(K)
C
      DATA (TS(K),K=1,NDPP)/240.0,245.0,250.0,255.0,260.0,
     Z265.0,270.0,275.0,280.0,285.0,290.0,295.0,300.0,305.0,
     Z310.0,315.0,320.0,325.0,330.0,335.0,340.0,345.0,350.0,
     Z355.0,360.0,365.0,369.75/
C
C     RHOGS:  SATURATED VAPOUR DENSITY (KG/CU.M)
C
      DATA (RHOGS(K),K=1,NDPP)/3.4425,4.144,4.95,5.8712,6.9191,
     Z8.1063,9.4467,10.9555,12.6496,14.5483,16.6729,19.048,21.702,
     Z24.6675,27.9838,31.6973,35.8665,40.5635,45.8841,51.9566,
     Z58.9620,67.1699,77.0107,89.2357,105.3784,129.6262,200.3185/
C
C     RHOFS:  SATURATED LIQUID DENSITY (KG/CU.M)
C
```

```
        DATA (RHOFS(K),K=1,NDPP)/570.1863,564.2021,558.1194,
       Z551.9287,545.6204,539.1833,532.6062,525.8726,518.9692,
       Z511.8777,504.5781,497.0466,489.2559,481.1731,472.7598,
       Z463.9683,454.7397,444.9993,434.6497,423.5608,411.55,398.3491,
       Z383.5352,366.3699,345.3413,316.2163,238.0881/
C
C       HFS:  SATURATED LIQUID SPECIFIC ENTHALPY(KJ/KG)
C
        DATA (HFS(K),K=1,NDPP)/442.06,453.74,465.57,477.55,489.69,
       Z501.98,514.44,527.07,539.87,552.86,566.05,579.46,593.1,
       Z606.99,621.17,635.55,650.47,665.69,681.36,697.55,714.37,
       Z731.95,750.51,770.43,792.49,818.95,871.09/
C
C       HGS:  SATURATED VAPOUR SPECIFIC ENTHALPY(KJ/KG)
C
        DATA (HGS(K),K=1,NDPP)/860.06,866.01,871.93,877.8,883.61,
       Z889.35,895.01,900.57,906.02,911.34,916.53,921.56,926.4,
       Z931.04,935.44,939.56,943.37,946.8,949.78,952.2,953.91,
       Z954.7,954.23,951.89,946.55,935.15,891.09/
        DO 3 I=1,NDPP
          VFS(I)=1.0/RHOFS(I)
          VGS(I)=1.0/RHOGS(I)
          VFG(I)=VGS(I)-VFS(I)
          HFG(I)=HGS(I)-HFS(I)
    3     CONTINUE
        RETURN
        END
C************************************************
        SUBROUTINE SUB(PC,HF,TS,VFS,HFS,VF,TF,DVDHPC)
C************************************************
C
C       THIS SUBROUTINE DETERMINES THE SUBCOOLED
C       THERMODYNAMIC PROPERTIES OF PROPANE USING
C       INPUTS OF PRESSURE AND ENTHALPY
C
        IMPLICIT REAL (A-H,O-Z)
        COMMON / SUBPP / PCE,HCE,DENSCE,TCE,NDPCI,NDPCJ
        DIMENSION HC(15),VC(15),TC(15),X(3,14,14,2),
     &            S(2,17),XX(3,2)
        DIMENSION DENSCE(14),PCE(14,14),TCE(14,14),HCE(14,14)
C         INPUTS FOR SUBROUTINE
          IINC=0
C
C         PRODUCE AN ISOBAR FOR ALL DENSITIES
C
          DO 5 I=1,NDPCI
            DO 6 J=1,NDPCJ
              IF((PCE(I,J)-PC).LE.0.0) THEN DO
                IF(J.EQ.NDPCJ) GO TO 10
                ELSE DO
                IF(J.EQ.1.AND.IINC.EQ.0) THEN DO
                  IINC=IINC+1
                  GO TO 11
                  ELSE DO
                IF(J.EQ.1.AND.IINC.NE.0) GO TO 10
                IF(J.EQ.NDPCJ) GO TO 7
                GO TO 77
                END IF
                END IF
 6            CONTINUE
```

```
            GO TO 77
 7          J=J-1
            JS=J-1
 77       P1=PCE(I,J-1)
          P2=PCE(I,J)
          P3=PCE(I,J+1)
          JS=J
          J=J-1
          DO 8 K=1,3
            X(K,I,J,1)=HCE(I,J)
            X(K,I,J,2)=TCE(I,J)
 8          J=J+1
          DO 9 L=1,2
            S(L,I)=XLAGR(P1,P2,P3,PC,X(1,I,JS-1,L),X(2,I,JS,L),
     &                   X(3,I,JS+1,L))
 9          CONTINUE
          HC(I)=S(1,I)
          TC(I)=S(2,I)
          VC(I)=1./DENSCE(I)
          GO TO 5
C
C         INPUT THE SAT'D PROPERTIES FOR INTERPOLATION
C
 11       HC(I)=HFS
          TC(I)=TS
          VC(I)=VFS
          GO TO 5
 10       HC(I)=TC(I)=VC(I)=0.0
 5        CONTINUE
 1      CONTINUE
C
C       INTERP. ALONG ISOBAR W.R.T. HF TO DETERMINE PROPERTIES
C
        DO 14 J=1,NDPCI
C
C         -0.001 IS USED DUE TO TOLERANCE FOR SAT'D PROP. INCONSISTANCY
C
          IF((HC(J)-HF).LE.-0.001)GO TO 14
          IF((HC(J)-HF).GT.-0.001)GO TO 15
 14       CONTINUE
 15       H1=HC(J-2)
          H2=HC(J-1)
          H3=HC(J)
          J=J-2
          DO 16 I=1,3
            XX(I,1)=TC(J)
            XX(I,2)=VC(J)
            J=J+1
 16         CONTINUE
            TF=XLAGR(H1,H2,H3,HF,XX(1,1),XX(2,1),XX(3,1))
            VF=XLAGR(H1,H2,H3,HF,XX(1,2),XX(2,2),XX(3,2))
            DVDHPC=DXLAGR(H1,H2,H3,HF,XX(1,2),XX(2,2),XX(3,2))
      RETURN
      END
C------------------------------------------------
      REAL FUNCTION VDPH*8(P1,HF,VF1,DELP,TS,VFS,HFS)
C------------------------------------------------
C     THIS FUNCTION EVALUATES DVDPHC ASSUMING LINEAR OVER A
C     SMALL PRESSURE RANGE OF DELP. THIS IS DONE BECAUSE DVDPHC
C     IS NOT INDEPENDENT OF P NEAR THE SATURATED LIQUID LINE.
```

```
C     DELP: DELTA P IN BAR
      IMPLICIT REAL(A-H,O-Z)
      P2=P1+DELP
      CALL BOIL(P2,TS2,HFS2,HGS2,HFG2,VFS2,VGS2,VFG2,
     &DHFDP2,DHGDP2,DVFDP2,DVGDP2)
      CALL SUB(P2,HF,TS2,VFS2,HFS2,VF2,TF2,DVDHP2)
      VDPH=(VF2-VF1)/DELP
      RETURN
      END
C****************************************
      SUBROUTINE SUP(PH,HG,TS,VGS,HGS,VG,TG,DVDHPH,DVDPHH)
C****************************************
C
C     THIS SUBROUTINE DETERMINES THE SUPERHEATED
C     THERMODYNAMIC PROPERTIES OF PROPANE USING
C     INPUTS OF PRESSURE AND ENTHALPY
C
      IMPLICIT REAL (A-H,O-Z)
      COMMON / SUPPP / PE,HHE,DENSHE,THE,NDPHI,NDPHJ
      DIMENSION DENSHE(17),PE(17,24),THE(17,24),HHE(17,24)
      DIMENSION HH(18),VH(18),TH(18),X(3,17,24,2),
     &          S(2,17),XX(3,2)
      NUM=NDPHI+2
      NUMM1=NUM-1
        TH(1)=HH(1)=VH(1)=0.0
        IINC=0
C
C     PRODUCE AN ISOBAR FOR DENSITIES
C
          DO 5 I=1,NDPHI
            DO 6 J=1,NDPHJ
              IF((PE(I,J)-PH).LE.0.0) THEN DO
                IF(J.EQ.NDPHJ) GO TO 10
                ELSE DO
                IF(J.EQ.1) GO TO 10
                  IINC=IINC+1
                IF(J.EQ.NDPHJ) GO TO 7
                GO TO 77
                END IF
 6            CONTINUE
              GO TO 77
 7            J=J-1
              JS=J-1
 77         P1=PE(I,J-1)
            P2=PE(I,J)
            P3=PE(I,J+1)
            JS=J
            J=J-1
            DO 8 K=1,3
              X(K,I,J,1)=HHE(I,J)
              X(K,I,J,2)=THE(I,J)
 8            J=J+1
            DO 9 L=1,2
              S(L,I)=XLAGR(P1,P2,P3,PH,X(1,I,JS-1,L),X(2,I,JS,L),
     &                    X(3,I,JS+1,L))
 9            CONTINUE
            HH(I+1)=S(1,I)
            TH(I+1)=S(2,I)
            VH(I+1)=1./DENSHE(I)
            IF(IINC.EQ.1)GO TO 13
          GO TO 5
 13       HH(I)=HGS
          TH(I)=TS
          VH(I)=VGS
          GO TO 5
 10       HH(I+1)=TH(I+1)=VH(I+1)=0.0
  5       CONTINUE
C
C     INTERP. ALONG ISOBAR W.R.T. HG TO DETERMINE PROPERTIES

        DO 14 J=1,NDPHI
          IF((HH(J)-HG).LE.0.0)GO TO 14
          IF((HH(J)-HG).GT.0.0)GO TO 15
 14       CONTINUE
 15       IF(HH(J+1).EQ.0.0.OR.J.EQ.NDPHI) J=J-1
          H1=HH(J-1)
          H2=HH(J)
          H3=HH(J+1)
          J=J-1
          DO 16 I=1,3
            XX(I,1)=TH(J)
            XX(I,2)=VH(J)
            J=J+1
 16         CONTINUE
            TG=XLAGR(H1,H2,H3,HG,XX(1,1),XX(2,1),XX(3,1))
            VG=XLAGR(H1,H2,H3,HG,XX(1,2),XX(2,2),XX(3,2))
            DVDHPH=DXLAGR(H1,H2,H3,HG,XX(1,2),XX(2,2),XX(3,2))
            CALL DERIH(PH,HG,DVDPHH)
      RETURN
      END
C------------------------------------
      SUBROUTINE DERIH(PI,HH,DVDPHH)
C------------------------------------
C
C     THIS SUBROUTINE DETERMINES THE PARTIAL DERIVATIVE
C     OF SPECIFIC VOLUME W.R.T. PRESSURE FOR SUPERHEATED
C     PROPANE
C
      IMPLICIT REAL (A-H,O-Z)
      COMMON / SUPPP / PE,HHE,DENSHE,THE,NDPHI,NDPHJ
      DIMENSION PH(18),VH(18),TH(18),X(3,17,24,2),
     &          S(2,17),XX(3,2)
      DIMENSION DENSHE(17),PE(17,24),THE(17,24),HHE(17,24)
      NUM=NDPHI+2
      NUMM1=NUM-1
        TH(1)=PH(1)=VH(1)=0.0
        IINC=0
C
C     PRODUCE AN ISENTHALPIC LINE FOR ALL DENSITIES
C
        DO 5 I=1,NDPHI
          DO 6 J=1,NDPHJ
            IF((HHE(I,J)-HH).LE.0.0) THEN DO
              IF(J.EQ.NDPHJ) GO TO 10
              ELSE DO
              IF(J.EQ.1) GO TO 10
              IINC=IINC+1
              IF(J.EQ.NDPHJ) GO TO 7
              GO TO 77
              END IF
 6            CONTINUE
              GO TO 77
 7            J=J-1
              JS=J-1
 77         H1=HHE(I,J-1)
            H2=HHE(I,J)
            H3=HHE(I,J+1)
            JS=J
            J=J-1
            DO 8 K=1,3
              X(K,I,J,1)=PE(I,J)
              X(K,I,J,2)=THE(I,J)
 8            J=J+1
            DO 9 L=1,2
              S(L,I)=XLAGR(H1,H2,H3,HH,X(1,I,JS-1,L),X(2,I,JS,L),
     &                      X(3,I,JS+1,L))
 9            CONTINUE
            PH(I+1)=S(1,I)
            TH(I+1)=S(2,I)
            VH(I+1)=1./DENSHE(I)
            IF(IINC.EQ.1) GO TO 13
            GO TO 5
C           THIS IS THE LIMIT OF THE LOWER ROOT OF HGS=HGS(P)
 13         IF(HH.GE.954.7) GO TO 10
            CALL BOILH(HH,PS,VHS,TS)
            PH(I)=PS
            TH(I)=TS
            VH(I)=VHS
            GO TO 5
 10         PH(I+1)=TH(I+1)=VH(I+1)=0.0
 5          CONTINUE
C
C     INTERP. ALONG ISENTHALPIC LINE W.R.T. TO DETERMINE DERIVATIVE
C
         DO 14 KK=1,NUMM1
           J=NDPHI-KK
           IF((PH(J)-PI).LE.0.0)GO TO 14
           IF((PH(J)-PI).GT.0.0)GO TO 15
 14        CONTINUE
 15        IF(PH(J-1).EQ.0.0.OR.J.EQ.1) J=J+1
           P1=PH(J-1)
           P2=PH(J)
           P3=PH(J+1)
           J=J-1
           DO 16 I=1,3
           XX(I,1)=TH(J)
           XX(I,2)=VH(J)
           J=J+1
 16        CONTINUE
           TG=XLAGR(P1,P2,P3,PI,XX(1,1),XX(2,1),XX(3,1))
           VG=XLAGR(P1,P2,P3,PI,XX(1,2),XX(2,2),XX(3,2))
           DVDPHH=DXLAGR(P1,P2,P3,PI,XX(1,2),XX(2,2),XX(3,2))
        RETURN
        END
C-----------------------------------
        SUBROUTINE BOILH(HH,P,VS,T)
C-----------------------------------
C       USING H AS AN INPUT DETERMINE THE SATURATED PROPERTIES
C       AT THAT H USING A LAGRANGIAN INTERPOLATION AVERAGE FIT
        IMPLICIT REAL (A-H,O-Z)
```

```
      COMMON / SATPP / TS,PS,HFS,HGS,VFS,VGS,NDPP
      DIMENSION TS(27),PS(27),HFS(27),HGS(27),VFS(27),VGS(27),
     &S(5),X(4,5),DS(5)
C
C     THIS PROGRAM LIMITS THE DVDPHH TO 30.0 BARS SINCE THERE ARE
C     2 ROOTS
C
      DO 1 J=1,NDPP
        IF((HGS(J)-HH).LE.0.0) GO TO 1
        IF((HGS(J)-HH).GT.0.0) GO TO 2
 1      CONTINUE
 2      H1=HGS(J-2)
        H2=HGS(J-1)
        H3=HGS(J)
        H4=HGS(J+1)
        J=J-2
        DO 3 I=1,4
          X(I,1)=TS(J)
          X(I,2)=PS(J)
          X(I,3)=VGS(J)
          J=J+1
 3        CONTINUE
C
C     BASED ON PRESSURE AT INITIAL TIME DETERMINE SATURATED
C     PROPERTIES USING LAGRANGIAN INTERPOLATION (AVERAGE OF
C     TWO LAGRANGIAN FITS)
C
      DO 4 J=1,3
        XV1=XLAGR(H1,H2,H3,HH,X(1,J),X(2,J),X(3,J))
        XV2=XLAGR(H2,H3,H4,HH,X(2,J),X(3,J),X(4,J))
        S(J)=(XV1+XV2)*0.5
 4      CONTINUE
      T=S(1)
      P=S(2)
      VS=S(3)
      RETURN
      END
C-----------------------------------------------------------------
      SUBROUTINE BOIL(P,T,HF,HG,HFG,VF,VG,VFG,DHFDP,DHGDP,DVFDP,DVGDP)
C-----------------------------------------------------------------
C     USING P AS AN INPUT DETERMINE THE SATURATED PROPERTIES
C     AT THAT P USING A LAGRANGIAN INTERPOLATION AVERAGE FIT
      IMPLICIT REAL (A-H,O-Z)
      COMMON / SATPP / TS,PS,HFS,HGS,VFS,VGS,NDPP
      DIMENSION TS(27),PS(27),HFS(27),HGS(27),VFS(27),VGS(27),
     &S(5),X(4,5),DS(5)
      DO 1 J=1,NDPP
        IF((PS(J)-P).LE.0.0) GO TO 1
        IF((PS(J)-P).GT.0.0) GO TO 2
 1      CONTINUE
 2      P1=PS(J-2)
        P2=PS(J-1)
        P3=PS(J)
        P4=PS(J+1)
C       JS=J
        J=J-2
        DO 3 I=1,4
          X(I,1)=TS(J)
          X(I,2)=HFS(J)
          X(I,3)=HGS(J)
          X(I,4)=VFS(J)
          X(I,5)=VGS(J)
          J=J+1
 3        CONTINUE
C
C     BASED ON PRESSURE AT INITIAL TIME DETERMINE SATURATED
C     PROPERTIES USING LAGRANGIAN INTERPOLATION (AVERAGE OF
C     TWO LAGRANGIAN FITS)
C
      DO 4 J=1,5
        XV1=XLAGR(P1,P2,P3,P,X(1,J),X(2,J),X(3,J))
        XV2=XLAGR(P2,P3,P4,P,X(2,J),X(3,J),X(4,J))
        S(J)=(XV1+XV2)*0.5
 4      CONTINUE
      T=S(1)
      HF=S(2)
      HG=S(3)
      VF=S(4)
      VG=S(5)
      HFG=HG-HF
      VFG=VG-VF
C
C     DETERMINE THE DERIVATIVES FROM THE LAGRANGE FITS
C
      DO 5 J=2,5
        XV3=DXLAGR(P1,P2,P3,P,X(1,J),X(2,J),X(3,J))
        XV4=DXLAGR(P2,P3,P4,P,X(2,J),X(3,J),X(4,J))
        DS(J)=(XV3+XV4)*0.5
 5      CONTINUE
        DHFDP=DS(2)
        DHGDP=DS(3)
        DVFDP=DS(4)
        DVGDP=DS(5)
      RETURN
      END
C---------------------------------------------------
      REAL FUNCTION XLAGR(C0,C1,C2,CX,U0,U1,U2)
C---------------------------------------------------
C
C     LAGRANGIAN INTERPOLATION
C
      IMPLICIT REAL (A-H,O-Z)
      XLAGR=(CX-C1)*(CX-C2)/(C0-C1)/(C0-C2)*U0-
     &       (CX-C0)*(CX-C2)/(C0-C1)/(C1-C2)*U1+
     &       (CX-C0)*(CX-C1)/(C0-C2)/(C1-C2)*U2
      RETURN
      END
C---------------------------------------------------
      REAL FUNCTION DXLAGR(C0,C1,C2,CX,U0,U1,U2)
C---------------------------------------------------
C
C     DERIVATIVE OF LAGRANGIAN INTERPOLATION
C
      IMPLICIT REAL (A-H,O-Z)
      A=(C0-C1)*(C0-C2)
      B=(C0-C1)*(C1-C2)
      C=(C0-C2)*(C1-C2)
      DXLAGR=U0*(2.0*CX-(C1+C2))/A-
     &       U1*(2.0*CX-(C0+C2))/B+
     &       U2*(2.0*CX-(C0+C1))/C
```

```
      RETURN
      END
C***********************************************************
C***********************************************************
C     PHENOMENOLOGICAL RELATIONS & TRANSPORT PROPERTIES
C***********************************************************
C***********************************************************
C---------------------------------------------
      REAL FUNCTION VENT(PIN,TG,PSHIGH,PSLOW,TDIAM)
C---------------------------------------------
C
C     THIS FUNCTION DETERMINES THE ISENTROPIC CHOKED
C     FLOW OF AN IDEAL GAS THROUGH THE P.R.V.
C
C     PIN=   PRESSURE INSIDE TANK (N/SQM)
C     TG=    VAPOUR TEMPERATURE (KELVIN)
C     DELTT= TIME STEP (SEC)
C     AW=    ATOMIC MASS OF PROPANE
C     GASCON=GAS CONSTANT (N*M/KG*K)
C     GAMMA= RATIO OF SPECIFIC HEATS(F.S.T.,PG.66)
C     TDIAM= DIAMETER OF THROAT (M);(F.S.T.,PG.66)
C     TAREA= THROAT AREA
C     TT=    TEMP @ THROAT (KELVIN
C     PT=    PRESS @ THROAT (N/SQM)
C     DENST= DENSITY @ THROAT (KG/M**3)
C     CD=    FLOW COEFFICIENT; (0.86;F.S.T.,PG.65)
C     FVENT= MASS VENTED OVER TIME STEP (KG)
C     PSL= LOW PRESSURE SET FOR VALVE TO CLOSE(PA)
C     PSH=HIGH PRESSURE SET FOR VALVE TO OPEN(PA)
C
      IMPLICIT REAL (A-H,O-Z)
      PIN=PIN*1.0E5
      PSH=PSHIGH*1.0E5
      PSL=PSLOW*1.0E5
      CD=0.86
      PI=3.141592654
      AW=44.0
      GASCON=8312.0/AW
      GAMMA=1.12
C
C     DETERMINE THE FLOW AREA AS A LINEAR FUNCTION OF PRESSURE
C
      AMAX=(PI*TDIAM**2.0)/4.0
      IF(PIN.GE.PSH) THEN DO
        TAREA=AMAX
        ELSE DO
        IF(PIN.LT.PSH.AND.PIN.GE.PSL)THEN DO
          TAREA=AMAX*(PIN-PSL)/(PSH-PSL)
          ELSE DO
          TAREA=0.0
          END IF
        END IF
      X=2.0/(GAMMA+1.)
C     TEMP. @ THROAT
      TT=X*TG
C     PRESS. @ THROAT
      PT=PIN*X**(GAMMA/(GAMMA-1.))
C     RATIO=PT/PIN
C     DENS. @ THROAT
      DENST=PT/(GASCON*TT)
      C=SQRT(GAMMA*GASCON*TT)
      VENT=CD*DENST*TAREA*C
      PRINT,'VENT=',VENT
      PIN=PIN*1.0E-5
      RETURN
      END
C-----------------------------------------------------------------------
      FUNCTION EVAP(AREA,TWALL,VFSM,QL,TS,TL,DELTT,HFG,ILEVEL)
C-----------------------------------------------------------------------
C
C     THIS SUBROUTINE DETERMINES THE MASS FLOW RATE
C     DUE TO EVAPORATION
C
C     AREA  =X-SECT. AREA OF INTERFACE IN M
C     TWALL =UPDATED ELEMENTAL INSIDE WALL TEMPS. IN K
C     TL    =LIQUID BULK TEMP. IN K
C     QL    =HEAT INPUT TO LIQUID SPACE IN KW
C     TWMEAN=MEAN LIQUID SPACE INTERNAL WALL TEMP. IN K
C     FILM  =FILM THICKNESS OF FROTHING AREA IN M
C     HTCINT=INTERFACE HEAT TRANSFER COEFF. IN KW/CU.M K
C     TFILM =TEMP. OF THE FILM IN K
C     HFG   =LATENT HEAT IN KJ/KG
      IMPLICIT REAL (A-H,O-Z)
      REAL KLIQ
      COMMON / INDEX / NBEG,NTOT,NANG
      COMMON / MESH / THETA,DELTH,DTHETA,R,DR,DELTR
      DIMENSION DR(15),DELTR(15),R(15),THETA(21),DELTH(21),TWALL(21,15)
C
C     DETERMINE MEAN INTERNAL WALL TEMP. IN LIQUID SPACE
C
      TWALT=0.0
      DENOM=0.0
      DO 1 I=ILEVEL,NANG
        TWALT=TWALT+DELTH(I)*TWALL(I,NBEG)
        DENOM=DENOM+DELTH(I)
 1      CONTINUE
      TWMEAN=TWALT/DENOM
      TFILM=(TWMEAN+TL)/2.0
      ROL=1./VFSM
      SPHL=CPSL(TFILM)/44.1
      KLIQ=CONDUC(TFILM,ROL)/1.0E+06
      VHD=4.0*QL/((TWMEAN-TFILM)*SPHL*ROL)
      VH=VHD*DELTT
      FILM=VH/(2.0*AREA)
      IF(FILM.EQ.0.0) GO TO 2
      HTCINT=KLIQ/FILM
      TM=TS
C     EVAPORATION RATE
      EVAP=AREA*HTCINT*(TFILM-TM)/HFG
      IF(EVAP.LT.0.0) GO TO 2
      GO TO 3
 2    EVAP=0.0
 3    RETURN
      END
C-------------------------------------------------------------
      REAL FUNCTION BUBBLE(VFS,VGS,EMF,XB,TAUB,TB,IOPEN)
C-------------------------------------------------------------
C
C     THIS SUBROUTINE DETERMINES THE MASS FLOW RATE
C     DUE TO BUBBLE RISE
```

```
C
C      XB    -LIQUID QUALITY
C      ALPHA -LIQUID VOID FRACTION
C      ROG   -DENSITY OF VAPOUR IN LIQUID SPACE (KG/M**3)
C      ROL   -DENSITY OF LIQUID IN LIQUID SPACE (KG/M**3)
C      G     -ACCELERATION OF GRAVITY (CM/SQS)
C      SIGMA -SURFACE TENSION OF PROPANE (G/SQS; DYNES/CM)
C      VELINF-BUBBLE TERMINAL VELOCITY (M/S)
C      MBUB  -MASS FLOW RATE OF BUBBLES (KG/S)
C      TAUB  -MEAN DISTANCE OF BUBBLE RISE (M)
C      COEFF -COEFFICIENT TO TAKE CARE OF LIQUID ENTRAINMENT(>=1)
C
       IMPLICIT REAL (A-H,O-Z)
C
C      DETERMINE VOID FRACTION ASSUMING NO SLIP
C
       RALPHA=1.0+((1.0-XB)/XB)*VFS/VGS
       ALPHA=1.0/RALPHA
C
C      DETERMINE DENSITY OF BUBBLES
C
       ROG=(1.0/VGS)/1000.0
       ROL=(1.0/VFS)/1000.0
       G=981.0
C
C      TAUB IS DETERMINED FROM LEVEL
C
C
C      DETERMINE TERMINAL VELOCITY FROM HARMATHY'S CORRELATION
C
       SIG=SIGMA(TB,
       VELINF=0.0153*(G*(ROL-ROG)*SIG/ROL**2.0)**0.25
C
C      DETERMINE MASS FLOW RATE
C
       IF(IOPEN.EQ.0) THEN DO
C      FOR OPEN VALVE
         COEFF=1.0
         ELSE DO
         COEFF=1.0
         END IF
       BUBBLE=COEFF*VELINF*EMF*XB/TAUB
C      PRINT,'SIG=',SIG,'ALPHA=',ALPHA,'VELINF=',VELINF,'BUB=',BUBBLE
       RETURN
       END
C
C---------------------------
       REAL FUNCTION SIGMA(TK)
C---------------------------
C      THIS FUNCTION EVALUATES THE SURFACE TENSION OF
C      PROPANE AS A FUNCTION OF TEMPERATURE (K).
C      THE INTERPOLATING FUNCTION IS DETERMINED FROM A
C      PROGRAM DEVELOPED BY  R.C. PRASAD
C      T      = TEMP IN CELSIUS
C      TK     = ABSOLUTE TEMP IN KELVIN
C      TCRIT  = CITICAL TEMP IN KELVIN
C      SIGMA  = SURFACE TENSION (DYNES/CM
       REAL*8 SIGO,MUE,B
       REAL TK,TCRIT,TEMPV
       SIGO=27.87011215
```

```
      MUE=0.8645988
      B=0.21334518
      TCRIT=369.8
C     DETERMINE THE SURFACE TENSION
      TEMPV=1.0-TK/TCRIT
      SIGMA=SIGO*(1.0+(B*TEMPV))*TEMPV**MUE
      RETURN
      END
C----------------------------
      FUNCTION VISCOS(T,RO)
C----------------------------
C     T = TEMPERATURE, K
      REAL*8 A,B,C,E,F,AJ1,AJ2,AJ3,AJ4,AJ5,AJ6,AJ7,GV(9)
      RO=RO*0.001
C
      A=0.0
      B=0.0
      C=1.12
      E=1.0
      F=358.9
      ROCR=0.211
C
      GV(1)=-0.1132608333D 07
      GV(2)=0.8465572858D 06
      GV(3)=-0.2314345388D 06
      GV(4)=0.2089338580D 05
      GV(5)=0.2693257510D 04
      GV(6)=-0.8709423636D 03
      GV(7)=0.9216623121D 02
      GV(8)=-0.4593654804D 01
      GV(9)=0.9088758027D-01
C
      AJ1=-0.1496346267D 02
      AJ2=0.2179858583D 02
      AJ3=-0.2566988314D 05
      AJ4=0.1527903946D 04
      AJ5=-0.2112678461D 00
      AJ6=0.2127695241D 03
      AJ7=0.5515987919D 05
C
      ROR=RO/ROCR
      THETA=ROR-1.0
C
C-----CALCULATE DILUTE GAS VISCOSITY (VO)
C
      VAR=T**(-4.0/3.0)
      VO=0.0
      DO 20 IH=1,9
        VAR=VAR*T**(1.0/3.0)
        VO=VO+GV(IH)*VAR
 20     CONTINUE
C
C-----CALCULATE MODERATE DENSITY CONTRIBUTION (V1)
C
      V1=(A+B*(C-DLOG(T/F))**2)*RO
C
C-----CALCULATE HIGH DENSITY CONTRIBUTION (V2)
C
      V2=E*DEXP(AJ1+AJ4/T)*(DEXP(RO**0.1*(AJ2+AJ3/T**1.5)+
     &   THETA*SQRT(RO)*(AJ5+AJ6/T+AJ7/(T*T)))-1.0)
```

```
C
C-----CALCULATE BACKGROUND VISCOSITY
C
      VHAN=V0+V1+V2
      VISCOS=VHAN
C
      RO=RO*1000.
      RETURN
      END
C---------------------------
      FUNCTION CONDUC(T,RO)
C---------------------------
C-----FUNCTION TO CALCULATE THERMAL CONDUCTIVITY,MILLIWATT/M-K
C     T= TEMPERATURE, K
C     RO=DENSITY, KG/M**3
C
      REAL*8 A,B,C,D,F,AK1,AK2,AK3,AK4,AK5,AK6,AK7,GT(9)
      RO=RO*0.001
C
      A=-1.149813131
      B=0.788531221
      C=1.12
      D=1.0
      F=358.9
      ROCR=0.211
C
      GT(1)=-0.1089381103D 07
      GT(2)=0.8343297829D 06
      GT(3)=-0.2270902736D 06
      GT(4)=0.1667866368D 05
      GT(5)=0.4347320565D 04
      GT(6)=-0.1177784671D 04
      GT(7)=0.1215425833D 03
      GT(8)=-0.6040596921D 01
      GT(9)=0.1207373681D 00
C
      AK1=-0.1092449464D 02
      AK2=0.1480622080D 02
      AK3=-0.1874097856D 05
      AK4=0.1269709514D 04
      AK5=-0.9598486895D-01
      AK6=0.1389024749D 03
      AK7=0.2554581504D 05
C
C-----CALCULATE THERMAL CONDUCTIVITY BY HANLEY'S CORRELATION
C
      ROR=RO/ROCR
      THETA=ROR-1.0
C
C-----CALCULATE DILUTE GAS THERMAL CONDUCTIVITY (TC0)
C
      VAR=T**(-4.0/3.0)
      TC0=0.0
      DO 20 IH=1,9
        VAR=VAR*T**(1.0/3.0)
        TC0=TC0+GT(IH)*VAR
 20     CONTINUE
C
C-----CALCULATE MODERATE DENSITY CONTRIBUTION (TC1)
C
```

```
      TC1=(A+B*(C-DLOG(T/F))**2)*RO
C
C     CALCULATE HIGH DENSITY CONTRIBUTION (TC2)
C
      TC2=D*DEXP(AK1+AK4/T)*(DEXP(RO**0.1*(AK2+AK3/T**1.5)+
     &    THETA*SQRT(RO)*(AK5+AK6/T+AK7/(T*T)))-1.0)
C
C-----CALCULATE BACKGROUND THERMAL CONDUCTIVITY
C
      TCHAN=TC0+TC1+TC2
      CONDUC=TCHAN
C
      RO=RO*1000.0
      RETURN
      END
C-------------------------
      REAL FUNCTION CPSL(T)
C-------------------------
C     SPECIFIC HEAT OF SATURATED LIQUID
C
      REAL EP,TCRT,A,B,C,D,ET,X,X2
C
C     DECLARING THE CONSTANTS
C
      EP=0.8
      TCRT=369.80
      A=4.201410
      B=79.737878
      C=18.313347
      D=-36.564448
      E=67.003225
C
C     CALCULATING THE SPECIFIC HEAT
C
      IF(T.LT.TCRT) GO TO 30
      CPSL=0.0
      RETURN
 30   X=T/TCRT
      X2=X*X
      Q=(1.0-X)**EP
      CPSL=A*X/Q+B+C*X+D*X2+E*X*X2
      RETURN
      END
C-----------------------------------------------------------------
      SUBROUTINE INCON(PIN,HFI,HGI,KSTATI,PERFIL)
C-----------------------------------------------------------------
      IMPLICIT REAL(A-H,O,Z)
      PIN=5.76
C     LIQUID ENTHALPY TO GET REQ'D INITIAL TEMP.
      HFI=511.0
C     VAPOUR ENTHALPY VERY CLOSE TO SATURATION
      HGI=907.
      KSTATI=1
      PERFIL=0.800
      RETURN
      END
C-----------------------------------------------------------------
      SUBROUTINE GEOM(IOPEN,PSHIGH,PSLOW,TDIAM,RIS,XLTH,DELTN,DELTV,
     &DTMIN,DTHETA,THCKS,VOLT,NSTEPS,TPRINT)
C-----------------------------------------------------------------
```

```
C     THIS SUBROUTINE IS USED TO SET THE GEOMETRY OF THE
C     TANK & VARIOUS CONSTANTS TO BE USED IN THE PROGRAM
C     IOPEN=1: PRV CLOSED
C          =0: PRV OPEN
C     PSHIGH: HIGH PRESSURE SET OF PRV IN BAR
C     PSLOW:  LOW PRESSURE SET OF PRV IN BAR
C     TDIAM:  VALVE THROAT DIAMETER IN M
C     THCKS:  THICKNESS OF SHELL IN M
C     RIS:    INSIDE RADIUS OF TANK SHELL IN M
C     DELTN:  NORMAL TIME STEPS IN SECONDS
C     DELTV:  TIME STEPS WHEN VALVE IS OPEN IN SECONDS
C     DTMIN:  MINIMUM TIME STEP IN SECONDS
C     DTHETA: ELEMENT ANGLES IN RADIANS
C     XLTH:     LENGTH OF TANK IN METRES
      IMPLICIT REAL(A-H,O,Z)
      NSTEPS=77
      PI=3.141592654
      DTHETA=PI/18.0
C
C     VALVE DATA
C
      IOPEN=1
      PSHIGH=19.2
      PSLOW=0.95*PSHIGH
C     FROM FULL SCALE REPORT; PAGE 66
      TDIAM=0.0178
C
C     SHELL DATA
C
      RIS=0.3048
      XLTH=3.556
      VOLT=PI*RIS**2.0*XLTH
      THCKS=0.00635
C
C     TIME DEFINITIONS
C
      TPRINT=20.
      DELTN=2.0
      DELTV=0.5
      DTMIN=0.5
      RETURN
      END
$ENTRY
```